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भाप टरबाइन के ऊष्मीय स्वीकरण परीक्षण के लिए विनियम

भाग 2 पद्धति ख — टरबाइनों के विभिन्न किस्मों तथा साइजों के लिए परिशुद्धता की व्यापक रेंज

Indian Standard

RULES FOR STEAM TURBINE THERMAL ACCEPTANCE TESTS

PART 2 METHOD B — WIDE RANGE OF ACCURACY FOR VARIOUS TYPES AND SIZES OF TURBINES

UDC 621·224·53·089·5

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BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
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Rotating Machinery Sectional Committee, ET 15

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Indian Standard

RULES FOR STEAM TURBINES THERMAL ACCEPTANCE TESTS

NATIONAL FOREWORD

This Indian Standard which is identical with IEC Pub 953-2 (1990), issued by the International Electrotechnical Commission, was adopted by the Bureau of Indian Standards on the recommendation of the Rotating Machinery Sectional Committee (ET 15) and approval of the Electrotechnical Division Council.

The text of IEC standard has been approved as suitable for publication as Indian Standard without deviations.

CROSS REFERENCES

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards which are to be substituted in their place are listed below along with their degree of equivalence for the editions indicated:

International Standard	*Corresponding Indian Standard*	*Degree of Equivalence*
ISO 31-3 : 1992 Quantities and Units — Part 3: Mechanics	IS 1890 (Part 3) : 1994 Quantities, identical units and symbols: Part 3 Mechanics (*first revision*)	Identical
IEC 953-1 (1990) Rules for steam turbine thermal acceptance tests — Part 1 : Method A — High accuracy large condensing steam turbines	IS 14198 (Part 1) : 1994 Rules for steam turbine thermal acceptance tests: Part 1 Method A — High accuracy for large condensing steam turbines	Identical

The concerned technical committee has reviewed the provisions of ISO 5167-1 : 1991 referred to in this adopted standard and has decided that it is acceptable for use in conjunction with this standard.

Only the English language text given in the International Standard has been retained while adopting it as Indian Standard, and as such the page numbers given here are not the same as in IEC publication.

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the results of a test, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

INTRODUCTION

The rapid development of measuring techniques, the increasing capacity of steam turbines and the introduction of nuclear power plants necessitated a revision of IEC Publication 46 (1962) regarding acceptance tests.

Since all the needs of the power industry in the different parts of the world could not be satisfied by one single publication, the complete standard is divided into two parts, describing two different approaches for conducting and evaluating thermal acceptance tests of steam turbines and which can be used separately:

a) Method A, which is Part 1 of the standard (IEC 953-1), deals with thermal acceptance tests with high accuracy for large condensing steam turbines.

b) Method B, which is Part 2 of the standard (IEC 953-2), deals with thermal acceptance tests with a wide range of accuracy for various types and sizes of steam turbines.

1) Basic philosophy and figures on uncertainty

Part 1 provides for very accurate testing of steam turbines to obtain the level of performance with minimum measuring uncertainty. The operating conditions during the test are stringent and compulsory.

Method A is based on the exclusive use of the most accurate calibrated instrumentation and the best measuring procedures currently available. The uncertainty of the test result is always sufficiently small that it normally need not be taken into account in the comparison between test result and guarantee value. This uncertainty will not be larger than about 0.3% for a fossil fired unit and 0.4% for a nuclear unit.

The cost for instrumentation and the efforts for preparing and conducting the tests will generally be justified economically for large and/or prototype units.

Method B provides for acceptance tests of steam turbines of various types and capacities with appropriate measuring uncertainty. Instrumentation and measuring procedures have to be chosen accordingly from a scope specified in the standard which is centred mainly on standardized instrumentation and procedures, but may extend eventually up to very high accuracy provisions requiring calibration. The resulting measuring uncertainty of the test result is then determined by calculating methods presented in the standard and normally, if not stated otherwise in the contract, taken into account in the comparison between test result and guarantee value. The total cost of an acceptance test can therefore be maintained in relationship with the economic value of the guarantee values to be ascertained.

The specifications of the operating conditions during the test are somewhat more flexible in this method; furthermore, procedures are recommended for treating cases where these specifications cannot be met.

When good-standardized instrumentation and procedures are applied in a test, the measuring uncertainty of the result will usually amount to 0.9% to 1.2% for a large fossil fuel fired condensing unit, to 1.1% to 1.4% for a nuclear unit and to 1.5% to 2.5% for back pressure, extraction and small condensing turbines. It is possible to reduce these values by additional improvement in instrumentation, primarily by additional measurements of primary mass flows and/or calibration of measuring devices for primary mass flow.

2) Main differences between Methods A and B

In Method A, much more detailed information concerning the preparation and conduct of the tests and the measuring techniques are contained for guidance of the parties to the test than in Method B. In Method B, the detailed treatment of the objectives is left somewhat more to the discretion and decisions of the participants and necessitates sufficient experience and expertise on their part.

3) Guiding principles

The requirements concerning the preparation and conditions of the test and especially such conditions of the test as duration, deviations and constancy of test conditions and acceptable differences between double measurements are more stringent in Method A.

The test should be conducted preferably within eight weeks after the beginning of the operation. It is the intent during this period to minimize performance deterioration and risk of damage to the turbine.

Preliminary tests including enthalpy drop tests should be made during this period to monitor HP and IP turbine section performance. However, these tests do not provide LP section performance and for this reason it is imperative to conduct the acceptance tests as soon as practicable.

Whatever the case, when using Method A, if an enthalpy drop test indicates a possible deterioration of the HP or IP section, or if the plant conditions require that the tests be postponed more than four months after the initial start, then the acceptance tests should be delayed.

An adjustment of the heat rate test results to start-up enthalpy drop efficiencies or for the effects of aging is not permitted when using Method A.

If the test has to be postponed, Method A proposes that the test be carried out after the first major internal inspection; several methods are proposed for establishing the approximate condition of the turbine prior to the tests.

4) Instruments and methods of measurement

a) Measurement of electrical power

In addition to the conditions required for the measurement of electric power, which are similar in both methods, Method A requires a check of the instruments by a comparison measurement after each test run; the permissible difference between double measurements is limited to 0.15%.

b) Flow measurement

For the measurement of main flows the use of calibrated pressure difference devices is required in Method A. The application of a device not covered by international standardization, the throat-tap nozzle, is recommended therein and details of design and application are given.

The calibration of these devices shall be conducted with the upstream and downstream piping and flow-straightener. Methods for the necessary extrapolation of the discharge coefficient from the calibration values are given.

In Method B standardized pressure-difference devices are normally applied for flow measurement. Calibration is recommended where a reduction of overall measuring uncertainty is desirable. Double or multiple measurement of primary flow is recommended for the reduction of measuring uncertainty and a method for checking the compatibility is described.

c) Pressure measurement
The requirements and recommendations for pressure measurements are essentially similar. Only the methods for the measurement of exhaust-pressure of condensing turbines differ to some extent.

d) Temperature measurement
The requirements are essentially similar in both methods. However detail requirements are more stringent in Method A:

- calibration before and after the test,
- double measurement of the main temperature with with 0.5 K maximum difference,
- thermocouples with continuous leads,
- required overall accuracy.

e) Steam quality measurements
Methods A and B are identical.

5) *Evaluation of tests*

The preparatory work for the evaluation and calculation of the test results is covered in a very similar manner in Methods A and B. However, quantitative requirements are more stringent in Method A.

Method B contains some proposals for handling cases where some requirements have not been met to avoid rejection of the test.

In addition, Method B contains detailed methods for calculation of measuring uncertainty values of measured variables and tests results.

Method B recommends other methods for conducting and evaluating of the tests after the specified period and without a previous inspection.

6) *Correction of test results and comparison with guarantees*

The correction of test results to guarantee conditions is covered in both Methods A and B.

Method A provides for the comparison of test results to guarantee without consideration of measuring uncertainty.

Method B gives a broader spectrum of correction procedures. Furthermore, the measuring uncertainty of the result is taken into account in the guarantee comparison.

7) *Proposals for application*

Since the acceptance test method to be applied has to be considered in the details of the plant design, it should be stated as early as possible, preferably in the turbine contract, which method will be used.

Method B can be applied to steam turbines of any type and any power. The desired measuring uncertainty should be decided upon sufficiently early, so that the necessary provisions can be included in the plant.

If the guarantee includes the complete power plant or large parts thereof, the relevant parts of either method can be applied for an acceptance test in accordance with the definition of the guarantee value.

1. Scope and object

1.1 *Scope*

The rules given in this standard are applicable to thermal acceptance tests covering a wide range of accuracy on steam turbines of every type, rating and application. Only the relevant portion of these rules will apply to any individual case.

The rules provide for the testing of turbines, whether operating with either superheated or saturated steam. They include measurements and procedures required to determine specific enthalpy within the moisture region and describe precautions necessary to permit testing while respecting radiological safety rules in nuclear plants.

Uniform rules for the preparation, carrying out, evaluation, comparison with guarantee and calculation of measuring uncertainty of acceptance tests are defined in this standard. Details of the conditions under which the acceptance test shall take place are included.

Should any complex or special case arise which is not covered by these rules, appropriate agreement shall be reached by manufacturer and purchaser before the contract is signed.

1.2 *Object*

The purpose of the thermal acceptance tests of steam turbines and turbine plants described in this standard is to verify guarantees given by the manufacturer of the plant concerning:

a) turbine plant thermal efficiency or heat rate;

b) turbine thermodynamic efficiency or steam rate or power output at specified steam flow conditions;

c) main steam flow capacity and/or maximum power output.

The guarantees with their provisions shall be formulated completely and without contradictions (see 2.4). The acceptance tests may also include such measurements as are necessary for corrections according to the conditions of the guarantee and checking of the results.

1.3 *Matters to be considered in the contract*

Some matters in these rules have to be considered at an early stage. Such matters are dealt with in the following sub-clauses:

Sub-clause	
1.1	(paragraph 4)
1.2	(paragraph 2)
3.1	(paragraph 3 and 4)
3.3.3	(paragraph 1)
6.6	
6.8	
6.9	(paragraph 1)

2. Units, symbols, terms and definitions

2.1 *General*

The International System of Units (SI) is used in these rules; all conversion factors can therefore be avoided.

The coherent units for all relevant quantities are given in the Table in 2.2. Some conversion factors are given as well for specific heat rates based on units other than W/W.

2.2 *Symbols, units*

For the purpose of these rules the following symbols, definitions and units shall be used:

Quantity	Symbol	Unit	Examples of multiples and sub-multiples	Other ISO units
Power	P	W	kW	
Flow rate	$\dot{m}$	kg/s		
Pressure, absolute	p_{abs}	Pa	kPa	bar[1]
Pressure, gauge	p_e	Pa	kPa	bar[1]
Ambient pressure (barometric)	p_{amb}	Pa	kPa	bar[1], mbar
Pressure difference	Δp	Pa	kPa	
Thermodynamic temperature	T, Θ	K		
Celsius temperature	t, θ			°C
Temperature difference	Δt	K		
Vertical distance	H	m	mm	
Specific enthalpy	h	J/kg	kJ/kg	
Specific enthalpy of saturated water	h'	J/kg	kJ/kg	
Specific enthalpy of saturated steam	h''	J/kg	kJ/kg	
Specific enthalpy drop	Δh	J/kg	kJ/kg	
Specific heat	c	J/kg·K	kJ/kg·K	
Quality, i. e., dryness fraction of saturated steam by weight	x	kg/kg	g/g	
Rotational speed	n	s^{-1}		min^{-1}
Velocity	v	m/s		
Density	ϱ	kg/m^3		
Specific volume	v	m^3/kg		
Diameter	D	m	mm	
Acceleration due to gravity	g	m/s^2		
Thermal efficiency	η_t	W/W	kW/kW	
Thermodynamic efficiency	η_{td}	W/W	kW/kW	
Heat rate	HR	W/W	kW/kW	kJ/kW·s, kJ/kW·h
Steam rate	SR	kg/W·s or kg/J	kg/kW·s kg/kJ	kg/kW·h
Heat flow rate	$\dot{Q}$	J/s	kJ/s	
Cavitation factor	K	1		
Concentration	C	According to nature of tracer		
Correction factor according to 6.6*a)*	F	1		
Correction factor according to 6.6*b)*	F^*	1		
Isentropic exponent	κ			
Discharge coefficient	C_d			
Flow coefficient	α			
General quantity	x[2]			
Weight factor for averaging	γ			
Confidence limit	V			
Relative measuring uncertainty of x	$\tau_x = \frac{V_x}{x}$			
Tolerance of steam table	R			

[1] Admitted by CIPM and ISO for temporary use with fluids only.

[2] According to application.

Relation between Heat Rate and thermal efficiency:

Units used for HR	*Relation*
W/W, kW/kW, kJ/kW · s	$HR = \dfrac{1}{\eta_t}$
kJ/kW · h	$HR = \dfrac{3600}{\eta_t}$
kJ/MW · s	$HR = \dfrac{1000}{\eta_t}$
kcal/kW · h	$HR = \dfrac{859.845}{\eta_t}$
BTU/kW · h	$HR = \dfrac{3412.14}{\eta_t}$

2.3 *Subscripts, superscripts and definitions*

Quantity	Subscript	Position or definition
Power	b	At generator terminals
	a	Taken by auxiliaries not driven by the turbine (see 4.2.3); (see also IEC 34)
	g	Net power output: $P_g = P_b - P_a$
	c	At turbine coupling, less power required by turbine auxiliaries, if driven separately (see 4.2.3)
	i	Internal to the turbine
	mech	Mechanical losses of pump and pump drive
Initial steam flow rate and output	max	Values for fully opened control valves
Steam condition and flow rate	1	Directly upstream of high pressure (HP) turbine stop valve(s) and the steam strainer(s) (if any) that are included in the turbine contract
	2	At exhaust of the turbine HP from which steam passes to the reheater
	3	Directly upstream of intermediate pressure IP turbine stop valves
	4	At exhaust of the turbine(s) discharging to the condenser
	e	At extraction point of extraction turbine
Condensate and feed water conditions and flow rates	5	At condenser discharge
	6	At inlet to condensate pump
	7	At discharge from condensate pump
	8	See Figure 1a
	9	At inlet of boiler feed pump
	10	At outlet of boiler feed pump
	11	At outlet of final feed heater
	b	After passage through the condensate pump and any coolers (oil, generator, gas/air) included in the contract
	d	At outlet from the drain cooler
	a	At outlet of air ejector condenser
	is	Refers to water taken from the feed-water system to the superheater for regulation of the initial steam temperature
	ir	Refers to water taken from the feed-water system to the reheater for control of the reheated steam temperature

Quantity	Subscript	Position or definition
Make-up water conditions and flow rate	m	Measurements adjacent to the inlet flange of the condensate system or of the evaporator
Gland steam conditions and flow rates	g	Steam supplied to glands from a separate source
	gl	Leak-off steam from glands and valve stems returned to the system and included in the measurement of the initial steam flow
	q	Flow of leak-off steam from glands and valve stems at inlet end or before a reheater which is led away for any extraneous purpose and neither it nor its heat is delivered to any part of the turbine cycle
	qy	Leak-off flows similar to q, but coming from a point or points downstream of a reheater
Main steam flow rate and concentration	M	Main steam flow at outlet of reactor
Mass flow rate and concentration	F	Refers to feed-water for reactor
	core	Refers to medium fluid passing through reactor core
	cond	Refers to condensed steam
	inj	Refers to injected tracer solution
	E	At entry into core of PWR
	R	Recirculated water flow from water separator
Condenser cooling water	w	
	wi	Condenser inlet
	wo	Condenser outlet
	wio	Average value between condenser inlet and outlet
Efficiency	t	Thermal
	td	Thermodynamic
Enthalpy drop	s	Refers to isentropic enthalpy drop
Velocity	throat	At throat of flow-metering nozzle
Static pressure	sat	Saturation pressure of water at pertinent temperature
Concentration	wat	In water phase
	L	In pump loop of BWR
	B	In blow-down water of PWR
	inj	Of injected tracer
	O	At injection point before tracer injection
Test results and guaranteed values	g	Guaranteed
	c	Corrected
	m	Measured
Correction Factor F or F^*	tot	Product of all individual correction factors
	1, 2, 3	Numbering of individual correction factors
	η	For correction of efficiency
	P	For correction of output
General use	i, j	Numbering subscripts

For confidence limit V and relative measuring uncertainty τ, subscripts which are identical to symbols for quantities always indicate the confidence limit applicable to the relative measuring uncertainty of this quantity.

Quantity	Superscript	Definition
Efficiency	/	Reference value of computer-calculated efficiency
General	–	Average value
	~	Weighted average value

2.4 *Definition of guarantee values and test results*

For the quantitative description of the thermodynamic performance of a steam turbine or steam turbine plant, several quantities are technically appropriate and generally applied. Guarantee values are expressed as such quantities and, consequently, test results are to be evaluated in the same manner.

The general definition of these quantities is always quite obvious. The details, however, may be different in each case and shall be fully considered (see also 1.2).

2.4.1 *Thermal efficiency*

For a power station turbine with regenerative feed heating, the thermal efficiency is the significant criterion. It is defined as the ratio of power output to heat added to the cycle from external sources.

$$\eta_t = \frac{P}{\Sigma (\dot{m}_j \, \Delta h_j)} \qquad (1)$$

where:

$\dot{m}_j$ are the mass-flows, to which heat is added

Δh_j are the resulting enthalpy rises

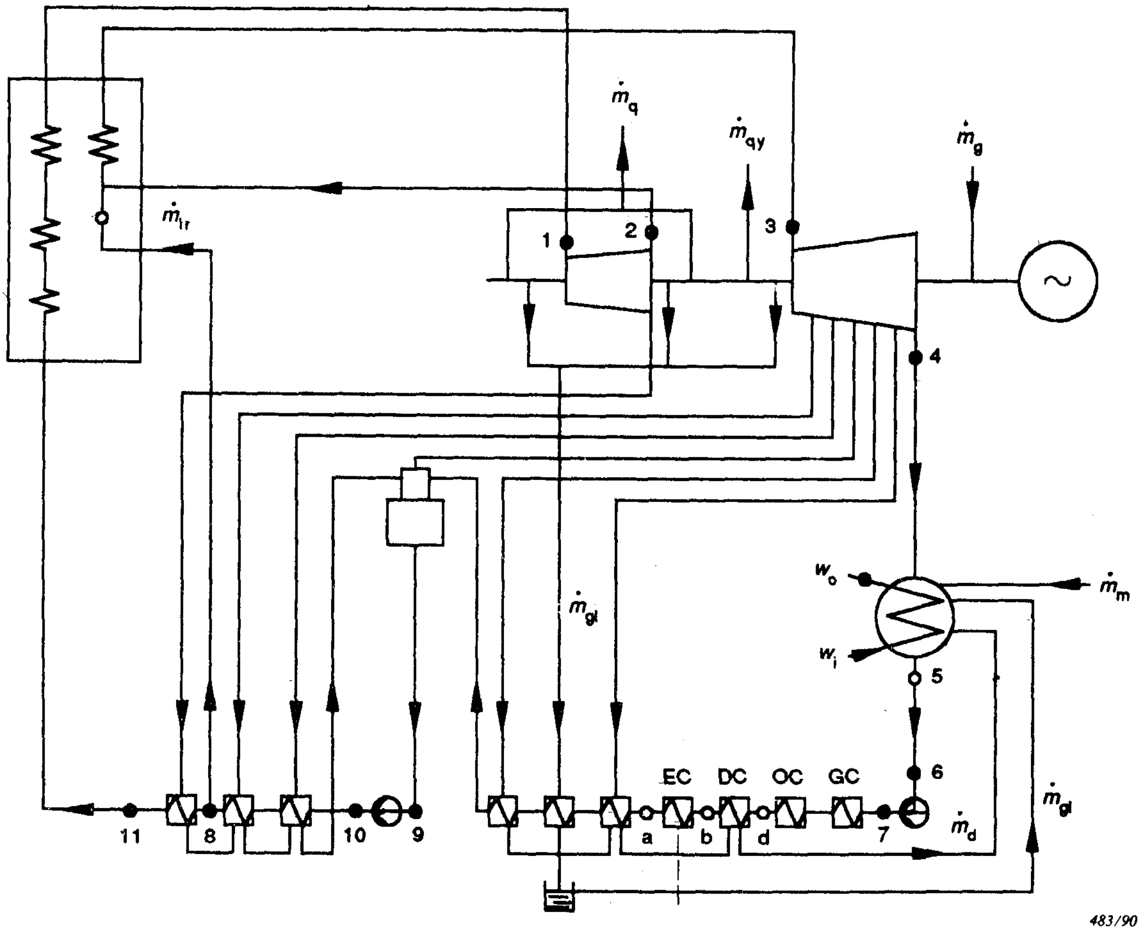
For each specific case a guarantee heat cycle, together with the guarantee terminal parameter, has to be defined as a basis for guarantee definition and test evaluation. It should be as simple as possible and as near as practicable to the cycle configuration to be realized for the test (see also 3.4.4).

A practical definition for a turbine plant with single reheat and feed heating according to Figure 1a is then:

$$\eta_t = \frac{P_b \text{ (or } P_g \text{ or } P_c\text{)*}}{\dot{m}_1 (h_1 - h_{11}) + \dot{m}_3 (h_3 - h_2)} \qquad (2)$$

---

* According to contract specification.



GC = generator gas cooler
OC = oil cooler
DC = drain cooler
EC = air ejector condenser

The point number remains the same for the same item of any other turbine type: for example, point 9 will be at the inlet of the feed pump, point 8 may be anywhere between downstream of point 6 and point 11.

FIG. 1a. – Reheating regenerative condensing turbine with feed-water heating.

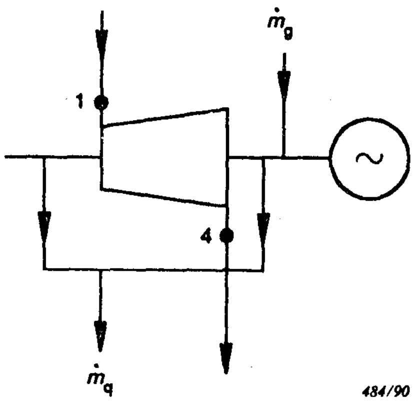


FIG. 1b. – Straight condensing or back-pressure turbine without feed-heating.

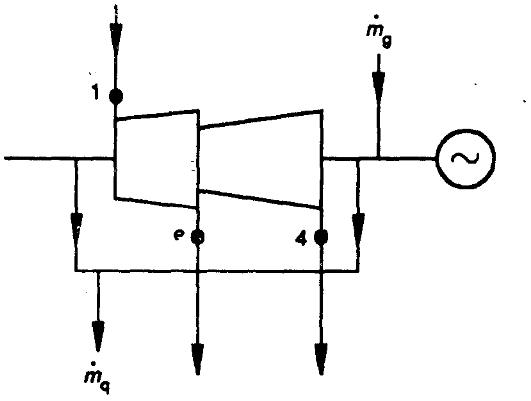


FIG. 1c. – Extraction condensing or back-pressure turbine without feed-heating.

FIG. 1. – Diagram for interpretation of symbols and subscripts.

Any additional heat and/or mass flows added to or subtracted from the cycle e.g., by make-up flow $\dot{m}_{m}$, spray attemperator flow m_{ir} or $\dot{m}_{is}$ or additional extraction for steam air preheater has to be accounted for in the evaluation by an appropriate correction of the test result (see Clause 6). Losses are not included in this definition, but treated according to 5.2.3.4.

To keep the sum of corrections small, it may be reasonable to include in the guarantee definition, by means of additional terms, important heat and mass flows present in the cycle configuration for the test for technical reasons (e.g., spray attemperator, reactor blow down, etc.).

This, however, also modifies the thermodynamic character of the definition and the resulting values of the thermal efficiency are not directly comparable with those according to formula (2). Furthermore, the correction procedure will not be avoided altogether in this way, because it is improbable that the values of these additional flows during the test will coincide exactly with those in the amended guarantee definition.

It is impracticable to describe in these rules all the possible variations in turbine cycles. In cases of complicated deviations of test cycle configuration from guarantee definition, it is advisable to use the correction procedures according to 6.6.1.

2.4.2 *Heat rate*

The heat rate traditionally has been used (and is still used) for the same objective as thermal efficiency, which is applied in these rules.

In a coherent unit system (SI):

$$HR = \frac{1}{\eta_t} \tag{3}$$

The unit of the so calculated heat rate is kW/kW = kJ/kW · s.

Heat rate values expressed in other units can be converted easily to thermal efficiency values, taking into account the appropriate conversion factors (see 2.2).

2.4.3 *Thermodynamic efficiency*

For a turbine receiving all the steam at one initial steam condition and discharging all the steam at a lower pressure (condensing or back-pressure turbine without regenerating feed-heating or reheat), the thermodynamic efficiency is the most appropriate measure of performance. It is defined as ratio of power output to isentropic power capacity (product of steam mass flow and isentropic enthalpy drop between initial steam condition and exhaust pressure):

$$\eta_{td} = \frac{P}{\dot{m}\,\Delta h_s} \tag{4}$$

The numerical value of thermodynamic efficiency does not depend on the initial steam and exhaust conditions, but is the indication for the efficiency of the expansion only.

The formula defining the thermodynamic efficiency for a straight condensing turbine without feed-heating according to Figure 1b, is then:

$$\eta_{\mathrm{td}} = \frac{P_{\mathrm{b}}\ (\text{or } P_{\mathrm{g}} \text{ or } P_{\mathrm{c}})^{*}}{\dot{m}_1 \cdot \Delta h_{\mathrm{s}\,1.4}} \tag{5}$$

where:

$\Delta h_{\mathrm{s}\,1.4}$ is the isentropic enthalpy drop between initial steam condition at point 1 and pressure at point 4

2.4.4 *Steam rate*

The steam rate traditionally has been used (and is still used) as a performance criterion for turbines as described in 2.4.3. It is defined as the ratio of initial steam flow rate to power output and is connected with thermodynamic efficiency as follows in coherent units (SI).

$$SR = \frac{\dot{m}}{P} = \frac{1}{\eta_{\mathrm{td}}\ \Delta h_{\mathrm{s}}} \tag{6}$$

Steam rate values expressed in other units can be converted to thermodynamic efficiency values after determination of the relevant Δh_{s} value taking into account the appropriate conversion factors (see 2.2).

Since numerical steam rate values depend also on initial steam and exhaust conditions, they are not comparable for turbines with different specified conditions. Therefore, thermodynamic efficiency is used in these rules.

2.4.5 *Main steam flow capacity*

The maximum flow rate of main steam with all regulating valves wide open under specified steam conditions (usually the steam conditions according to the definition of the other guarantee values) is the measure of the maximum flow capacity of the turbine.

2.4.6 *Maximum power output*

The power output of the turbine at the maximum flow rate of main steam can be guaranteed for a specified heat-cycle, which may differ to some extent from the guarantee heat cycle valid for the thermal efficiency. A maximum power output guarantee can be relevant for district heating turbines, etc.

2.4.7 *Guarantee values for extraction and mixed-pressure turbines***

For mixed-pressure turbines and single or multiple-extraction turbines with or without regulated extraction pressure(s), in all cases without regenerative feed-heating, the guarantee values are most appropriately defined as thermodynamic efficiency in the following generalized way:

$$\eta_{td} = \frac{P}{\Sigma\,(\dot{m}_{\mathrm{j}} \cdot \Delta h_{\mathrm{sj}})} \tag{7}$$

where $\dot{m}_{\mathrm{j}}$ and Δh_{sj} are the steam flow rate and isentropic enthalpy drop respectively for each consecutive section of the turbine.

---

* According to contract specification.

** Mixed pressure turbines – several inlets at different pressures during the expansion.

For example, for a single extraction turbine (either back-pressure or condensing), according to Figure 1c, this results in:

$$\eta_{td} = \frac{P_b \text{ (or } P_g \text{ or } P_c)^*}{\dot{m}_1 \cdot \Delta h_{s\,1,e} + (\dot{m}_1 - \dot{m}_e)\,\Delta h_{s\,e,4}} \qquad (8)$$

In the case of an extraction or mixed-pressure turbine, however, it is necessary to specify in addition to the values of output, initial steam flow rate and thermodynamic efficiency, the pertinent flow rate value(s) of extraction steam flow(s) and secondary steam flow(s). This applies at each load subject to guarantee.

Guarantee values for mixed-pressure or extraction turbines with regenerative feed-heating can be defined, if more practicable, as thermal efficiency also. In this case, too, the pertinent flow rate value(s) of secondary steam and extraction steam flow(s) is (are) necessary for a complete guarantee definition.

3. Guiding principles

3.1 *Advance planning for test*

The parties to any test under these rules shall reach agreement, when the plant is being designed, as to the testing procedure, the interpretation of guarantees, the number, location and arrangement of measuring points and measuring devices, valves and piping arrangements. This especially applies to steam turbines in nuclear power stations where subsequent modifications are often impracticable and the measuring points are not always accessible once the plant has started to operate. It is recommended that for the most important measurements special connection facilities such as flanges and thermometer wells be provided for the measuring equipment so that the acceptance tests can be carried out without impairing the instruments for normal operation.

The instrumentation has to be selected in such a way that power and heat flows which enter and leave the "system", as defined in the contract, and the conditions at its boundaries, can be determined.

The parties interested in the test shall reach agreement at a date as early as practicable on the measuring accuracy desired for the acceptance tests. This accuracy, with due consideration of the cost of the acceptance test, shall be in reasonable relationship with the significance of the plant.

The following is a list of typical items upon which agreement should be reached during the plant design:

a) Location of, and piping arrangement around, flow measuring devices on which test calculations are to be based.

b) Number and location of valves required to ensure as far as practicable that no unaccounted for flow enters or leaves the test cycle or bypasses any cycle component.

c) Number and location of temperature wells and pressure connections required to ensure correct measurements at critical points.

d) Number and location of duplicate instrument connections required to ensure correct measurements at critical points.

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* According to contract specification.

e) Handling of leak-off flows to avoid complications in testing or the introduction of errors.

f) Means of measuring pump shaft leakages, if necessary.

g) Method of determining steam quality, including sampling technique as required. The recommended methods are given in 4.7.

3.2 *Preparatory agreements and arrangements for tests*

a) Before the test the parties to the tests shall reach an agreement as to the test programme, the specific objectives of the tests, the measuring methods and the method of operation with due regard to limitation of the necessary corrections, the method for correcting the test results and for guarantee comparison with due regard to the contract conditions.

b) Agreement shall be reached as to the variables to be measured, the measuring instruments and who is to supply them, the location of the indicating instruments and the operating and recording personnel required.

c) Agreement shall be reached on the methods of obtaining the comparison measurements (see 3.5).

d) Agreement shall be reached on such matters as the means of securing constancy of steam conditions and output.

e) Instruments liable to failure or breakage in service should be duplicated by reserve instruments, properly calibrated, which can be put into service without delay. A record of such change of instrument during a test shall be clearly made on the observer's record sheet.

Instruments shall be located and arranged so that they may be read accurately with comfort by the observer. The calibration environment should be as close as practicable to the environment in which the instrument will operate during the test. This may be accomplished by locating the instruments in a controlled environment.

f) The determination of the enthalpy of steam superheated less than 15 K, or of the quality of steam containing moisture, may be made only when the parties to the test agree upon the method to be employed for this determination. The agreement, the method for making the determination and the method of applying the enthalpy or quality values to the test results shall be fully described in the test report.

The rate of flow of steam of any quality may be determined, where practicable, by condensing it completely and then measuring the condensate flow rate.

g) Agreement shall be reached as to the method of calibration of instruments, and by whom and when.

h) For any of the measurements necessary for a test under these rules, any methods may be employed other than those prescribed in these rules, provided they are mutually agreed upon in writing before the test by the parties to the test. Any such departure from the prescribed methods shall be clearly described in the test report. In the absence of written agreements, the rules herein shall be mandatory.

i) An independent expert may be a party to all agreements.

j) Agreement shall be reached on the minimum number of operating and recording personnel that is required.

3.3 *Planning of the test*

3.3.1 *Time for acceptance tests*

The acceptance tests shall be performed at the operating plant or, by mutual agreement, on the test bed of the manufacturer. Acceptance tests on site shall be carried out , if practicable, within eight weeks* after the first synchronization (see 3.5.). In any event, except with written agreement to the contrary, the acceptance test shall take place within the guarantee period specified in the contract.

3.3.2 *Direction of acceptance tests*

The responsibility for the direction of the acceptance tests shall be clearly assigned by the parties prior to the test, preferably to one person. This person is responsible for the correct execution and evaluation of the acceptance tests, and serves as arbitrator in the event of disputes as to the accuracy of observations, conditions, or methods of operation. He is entitled and obliged to obtain information on all necessary details.

Accredited representatives of the purchaser and the manufacturer may at all times be present to verify that the tests are conducted in accordance with these rules and the agreements made prior to the tests.

A party to the contract who is not responsible for the direction of the acceptance tests shall also be given the opportunity of obtaining information a sufficient time before the tests.

3.3.3 *Cost of acceptance tests*

The contract shall stipulate who is to bear the costs of acceptance tests and any repeated acceptance tests (see also 3.5, 3.7 and 3.9).

3.4 *Preparation of the tests*

3.4.1 *Condition of the plant*

Prior to commencement of the acceptance tests, it is essential to be satisfied that the steam turbine and driven machine are in suitable condition, together with the condenser and/or feedheaters if included in the guarantee. It is also essential to verify that leaks in the condensers, feedheaters, pipes and valves have been eliminated.

Prior to the acceptance tests, the supplier shall be given the opportunity to check the condition of the plant, if necessary by making his own measurements. Any deficiency determined at this time shall be rectified.

Although these rules deal specifically with the performance testing of steam turbine generators, it is required that all other equipment supplied as part of the same turbine generator contract shall be in full and correct working order and in normal commercial disposition during the turbine generator tests. This requirement does not apply if other equipment has been ordered as an extra to the contract after the performance guarantees have been contractually agreed, or if special measures to render the equipment non-operative during the tests have been agreed beforehand between the parties to the test and are described in prominent detail in the test report. An example of equipment in this category would be piping and valves, supplied as part of the same turbine generator contract, designed to permit steam to bypass part or all of the turbine expansion stages for temperature-matching purposes during start-up.

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* It is the intent during this period to minimize performance deterioration and risk of damage to the turbine. Enthalpy drop tests or preliminary tests should be made during this period to monitor HP and IP turbine section performance. However, these tests do not provide LP section performance and for this reason it is imperative to conduct the acceptance tests as soon as practicable.

3.4.2 *Condition of the steam turbine*

The condition of the steam turbine depends on the effects of ageing[1] (see 6.9), of partial damage[2] and of deposits[3].

The condition of the steam turbine can be determined by an internal inspection of the steam path generally by opening the steam turbine or by comparison measurements according to 3.5.

The absence of partial damage and/or deposits shall be confirmed before the test (see also 3.4.1).

Opening the turbine or an individual casing in order to locate a defect may be taken into consideration if large and inexplicable deviations are apparent from the comparison measurements.

3.4.3 *Condition of the condenser*

When the guarantees include the condenser performance and are based upon cooling water flow and temperature, the condenser shall be clean and the system shall be tested for sufficient air tightness. Agreement of the parties regarding these matters shall be reached.

The condition of the condenser shall be checked by opening the water boxes or measuring the terminal temperature difference. In the case of deposits[3], the condenser shall be cleaned by the purchaser prior to the acceptance tests at the request of the supplier or the parties concerned by the test may agree on a suitable correction.

3.4.4 *Isolation of the cycle*

The accuracy of test results depends to a large extent on effective isolation of the system. Extraneous flows should be isolated from the system and internal flows which bypass in an unaccounted-for way either cycle components or mass flow measuring equipment should be eliminated, if practicable, to obviate the need for measurement. If there is any doubt about the ability to isolate these flows during the test, preparations shall be made prior to the test to measure them.

All unused connections shall be blanked-off. If this is not practicable, the connections shall be broken at a suitable point so that the outlets are under constant observation.

The equipment and flows to be isolated and the methods to accomplish this should be agreed well ahead of the initial operation date of the turbine. The isolation of the cycle has to be described in the test report.

When the system is properly isolated for a performance test, the unaccounted-for leakage expressed in percent of initial steam flow at full load should not exceed the relative measuring uncertainty of the test result (expressed in percent) multiplied by 0.4 (see 5.2.3.4). Water storage in the condenser hotwell, deaerating and other extraction feedwater heaters, boiler drum(s), moisture separators, reheaters, and any other storage points within the cycle are to be taken into account (see also 5.2.3.4).

3.4.4.1 *Isolation of equipment and flows*

The following list includes items of the equipment and extraneous flows that should be isolated from the primary turbine feedwater cycle, if practicable:

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[1] Ageing means deterioration during correct operating and shutdown conditions in a manner such that the performance of the turbine is adversely influenced.

[2] Partial damage: damage and variations in the clearance of the blading, of balance pistons and shaft glands, valve spindle wear; leakages in the valve glands and seats.

[3] Deposits: salts or metal on the steam side, slime, ashes, bacteria, algae, etc., on the cooling water side.

a) Large-capacity storage tanks.

b) Evaporators and allied equipment such as evaporator condenser and evaporator pre-heaters.

c) Bypass systems and auxiliary steam lines for starting, if compatible with safe operation.

d) Bypass lines for condensate primary flow-measuring devices.

e) Turbine sprays.

f) Drain lines on stop, intercept and control valves.

g) Interconnecting lines to other units.

h) Demineralizing equipment.

Isolation of demineralizing equipment does not necessarily mean removing the equipment from the cycle. It does, however, mean that all ties with other units shall be isolated and such equipment as recirculating lines that affect the primary flow measurement shall be isolated or the flows measured.

i) Chemical-feed equipment using condensate.

j) Boiler vents.

k) Steam-operated soot blowers.

l) Condensate and feedwater flow bypassing heaters.

m) Heater drain bypasses.

n) Heater shell drains.

o) Heater water-box vents.

p) Hogging jets.

q) Condenser water-box priming jets.

r) Steam or water for station heating.

s) Blow-down from steam generator.

3.4.4.2 *Flows which shall be determined if not isolated*

The following extraneous flows which enter or leave the cycle in such a manner as to cause an error in the flows through the turbine shall be measured or estimated, if they are not isolated from the system.

a) Boiler fire door cooling flow and boiler-slag-tap cooling coil flow.

b) Sealing and gland cooling flow on the following (both supply and return):

  1) condensate pumps;
  2) boiler feed pumps;
  3) boiler-water or reactor circulating pump;
  4) heater drain pumps when not self-sealed;

  5) turbines for turbine-drive pumps;
  6) control rod seals on nuclear reactors.

c) Desuperheating water.

d) Boiler feed pump minimum-flow lines and balance drum flow.

e) Steam for fuel oil atomization and heating.

f) Boiler blowdowns.
g) Boiler fill lines.
h) Turbine water-seal flow.
i) Desuperheating water for turbine cooling steam.
j) Emergency blowdown valve of turbine gland leakage and sealing system.
k) Turbine water-seal overflows.
l) Steam and water lines installed for washing the turbine.
m) Steam, other than gland leak-off steam, to the steam-seal regulating valve.

n) Make-up water, if necessary.
o) Pegging steam (such as higher stage extraction at light loads) for low-pressure operation of deaerator.
p) Heater vents shall be closed if practicable. If not, they shall be throttled to a minimum.

q) Deaerator overflow line.
r) Water leakage into any water-sealed flanges, such as water-sealed vacuum breakers.
s) Pump seal leakage leaving the cycle.
t) Automatic extraction steam for industrial uses.
u) Steam to air preheaters (if isolation is not practicable).
v) Water and steam sampling equipment. If it is impracticable to isolate water and steam sampling equipment, and if the sampling flow is significant, it shall be measured.

w) Deaerator vents.
x) Reactor-core spray.
y) Sub-cooled water used for moisture separator or reheater coil drain cooling.

z) Continuous drains from wet steam turbine casings and connection lines (if not included in the guarantee).

3.4.4.3 *Suggested methods and/or devices for isolating equipment from turbine feedwater cycle*

The following methods are suggested for isolating miscellaneous equipment and extraneous flows from the primary turbine feedwater cycle and for checking such isolation:

a) Double valves and tell-tales.
b) Blank flanges.
c) Blank between two flanges.
d) Removal of spool piece for visual inspection.
e) Visual inspection for steam blowing to atmosphere – such as safety valves.

f) Closed valve which is known to be leak-proof (test witnessed by both parties) and not operated prior to or during the test.
g) Temperature indication (only acceptable under certain conditions – mutual agreement necessary).
h) Accurate measurements of levels in any tanks which should be isolated from the system.

i) Very important isolating valves (for example, HP and LP bypass valves) should be inspected and, if necessary, sealed prior to the test.

3.4.5 *Checks for leakage of condenser and feedwater heaters*

Condensers and feedwater heaters shall be checked for leakage and steps shall be taken to eliminate any significant leakage (see Appendix A).

In case of doubt, the check can be repeated after the test.

3.4.6 *Cleanliness of the steam strainers*

If necessary, the steam strainers shall be cleaned before the acceptance tests.

3.4.7 *Checking of the test measuring equipment*

All measuring equipment shall be examined before the test as to its condition and suitability. It shall be further established whether the measuring instrument, the point of installation, and the installation are in accordance with relevant requirements. The results of all these checks shall be recorded in the test report.

3.5 *Comparison measurements*

For comparison measurements, only those variables are taken into account which are essential to establish the condition of the steam turbine. For the comparison, it is advisable to measure the internal efficiency, the pressure and temperature at the extractions, the gland leakage flows, and the condenser terminal difference as well as the vibration level of the steam turbine rotor at suitable points.

The measuring uncertainty of comparison measurements should not be larger than that of the acceptance tests.

The type and scope of the comparison measurements as well as the costs shall be agreed upon by the purchaser and the supplier (see 3.3.3).

The parties interested in the acceptance test shall participate in the comparison measurements.

The reference measurement shall be made immediately after the first start-up, at partial load if necessary. Check measurements may be made:

a) before the acceptance test or during the preliminary test;

b) after blade washing;

c) before and/or after an inspection.

The measurements taken during the acceptance test can normally be evaluated as a final check measurement.

Should the result of a check measurement point towards a deposit which can be removed by blade washing, the supplier may require of the purchaser that the steam turbine be washed. If the check measurement sufficiently agrees with the reference measurement after the washing, the acceptance test may be carried out. Should the check measurement not agree sufficiently with the reference measurement, the parties to the contract should then decide either to eliminate the defect or to carry out the acceptance test. If the tests proceed, the comparison measurements shall be included in the test report.

If the result of the reference measurement differs considerably from the expected values according to the guarantee, remedial measures may be agreed upon by the purchaser and the supplier.

3.6 *Settings for tests*

3.6.1 *Load settings*

The acceptance tests for a turbine with partial arc admission shall be conducted at "valve points", which are those positions of the control valves where the total throttle effects are minimized.

The output at a "valve point" may only deviate from the specified value within the limits stated in 3.8.2.

In addition, the position of the LP control valves of extraction turbines should be brought to a "valve point", if practicable, by adjusting the extraneous steam flow. Similarly, the "valve points" of the secondary initial steam control valves of double pressure turbines can be achieved by adjustment of the secondary steam flow, if practicable.

These adjustments may result in deviations of the output from the specified value greater than stated in 3.8. In these cases a re-calculation of the guarantee value according to 6.6.1 may be made by agreement of the parties.

If large fluctuations in output or steam flow due to operating conditions are occurring, it is permitted to use a device limiting further opening of valves in order to keep optimum valve positions. The relative positions of the valves shall not differ throughout the test period.

In the event of variable pressure operation of a full arc admission turbine, tests shall be conducted with control valves wide open. The same applies to throttle control if the guarantee is based upon wide open control valves.

3.6.2 *Special settings*

No special adjustments shall be made to the turbine for test purposes that are inappropriate for commercial and continuous operation under any and all of the specified outputs and operating conditions. Exceptions are air bleed adjustments to control the vacuum, the use of load-limiting devices or other similar test control means, such as isolation of the cycle to prevent extraneous flows and/or internal by-pass flows of steam or water by closing certain drains or other valves. These shall be consistent with the terms of the guarantee and practicable with regard to operating safety and technique.

The turbine shaft seals shall be adjusted to normal operating condition before a test and arrangements made to measure outward or inward flow that will influence test results.

3.7 *Preliminary tests*

Preliminary tests may be run for the purpose of

a) determining whether the turbine is in a suitable condition for the conduct of an acceptance test,

b) checking all instruments,

c) training personnel in test procedures.

After a preliminary test has been made, it may be considered, if mutually agreed, as an acceptance test.

If the preliminary tests are unsatisfactory, the causes shall be investigated and, if necessary, the turbine shall be placed at the disposal of the manufacturer so that the latter may examine it and ascertain whether it is in a suitable condition for an acceptance test.

3.8 *Acceptance tests*

3.8.1 *Constancy of test conditions*

All tests shall be preceded by a period of temperature and flow stabilization, the duration of which shall be agreed upon by the parties interested in the test, since it will vary with the size of the turbine, the internal conditions and the amount of load change.

Any condition, the variation of which may influence the results of the test, shall be caused to become as nearly constant as practicable before the test begins, and shall be so maintained throughout the test within the permissible variations according to 3.8.2.

In order to maintain constancy of throttling, the travel of the governing valve, in the *opening* direction only, should be limited at the chosen position and the governor should be sufficiently overwound to ensure that it does not respond to normal changes in system frequency.

3.8.2 *Maximum deviation and fluctuation in test conditions*

Except upon agreement to the contrary by the parties, the maximum permissible deviation of the average test condition for each variable from that specified and the maximum permissible fluctuation of the variable during any one test run shall not exceed the limit prescribed in Table I.

Should these requirements not be met, the measurement may serve as information only, unless special agreement relating to the influence of the deviating operating conditions is reached.

3.8.3 *Duration of test runs and frequency of reading*

The requisite test duration is dependent on the steadiness of operating conditions and the speed of acquisition of test data. Accurately measurable changes in level of water stored in the system may be a limiting factor.

The recommended minimum test period (see 5.1) of an acceptance test is 1 h. This may be reduced by agreement or technical necessity, but should be not less than 30 min. The duration of the capacity tests should be agreed between the parties, but should be not less than 15 min.

Related test data recorded with indicating measuring equipment shall, if practicable, be read at the same time. Since the values measured are not constant, random errors cannot be avoided when taking readings at regular time intervals. The intervals shall be sufficiently short to ensure that these errors do not substantially affect the measuring uncertainty of the whole. This is of vital importance for readings of differential pressure for mass flow measurement and electrical output. For tests of one hour's duration readings at one-minute intervals will satisfactorily fulfil requirements for maximum deviations allowed in Table I. Pressures and temperatures for the determination of thermodynamic properties may be read at longer intervals, that is, from 3 min to 5 min according to amount and type of fluctuations.

If the adoption of longer intervals between readings is necessary a correspondingly longer duration of the test may be required.

The time of the reading should be indicated to the observers through signals from a main clock. It may alternatively be taken from the observers' watches, which shall be synchronized with each other prior to each test.

TABLE I

Maximum deviations and fluctuations[1] *in operating conditions*

Variable	Maximum permissible deviation of average of test from that specified
Initial steam pressure	±5%[2]
Initial steam temperature	±15 K[2]
Dryness fraction	±0.005
Extraction pressure (regulated)	±5%[2]
Exhaust pressure:	
- for feed heating	See [4]
- for back pressure turbines	±5%[2]
- for condensing turbines	±25% if condenser is not included in guarantee
Extraction flow rate	±10%
Reheat temperature	±15 K
Isentropic enthalpy drop	±7%
Output or initial steam flow	±5% after correction for specified conditions
Cooling water flow	±15% } if condenser is included in guarantee
Cooling water inlet temperature	±5 K
Final feed water temperature	±10 K
Speed	±2%[3]

[1] The maximum fluctuation of the variable permitted during each test shall not be greater than half the permissible deviation given in Table I except for the output which may vary by ±3%.

[2] All these items may not lead to a deviation of enthalpy drop of more than ±7%.

[3] If permitted by the technical guarantee of the turbine.

[4] Reasonably small deviations in the extraction pressures as compared with the design values normally have a negligible effect on the overall performance. Should there be disproportionately large deviations in the extraction steam flows suggesting malfunctioning of the heaters, the effect on the overall performance may be serious, and agreement shall then be reached on the course to be followed.

3.8.4 *Reading of integrating measuring instruments*

The mean values of the electrical output and the mass flow can also be determined by integrating measuring instruments by dividing the difference in the readings at the beginning and at the end of a test by the corresponding time interval.

All integrating measurement instruments should be read simultaneously. Related indicating measurement instruments should be read at or nearly at the same time.

It is advisable to take simultaneous readings of all integrating instruments at regular intervals during the test. This will enable checks for consistency to be made and permits the period for evaluation to be adjusted, if required, after the conclusion of the test.

All observations should begin somewhat before and stop somewhat after the intended test period provided all operating conditions remain constant.

3.8.5 *Alternative methods*

Alternative methods are presented in these rules for conducting certain details of the test. The test report shall state which alternative has been employed.

3.8.6 *Recording of tests*

Each observer shall record his actual observations. All records shall be made with carbon copies, at least in duplicate, or by mutual agreement be photo-copied immediately after each test-run. Immediately after the test each interested party shall receive a complete set of records.

3.8.7 *Additional measurement*

Should a deficiency be found during a test, which is rectified within a relatively short time, the test may be continued. In this event supplementary measurements shall be taken, if necessary, provided always that the correction involved can be calculated sufficiently accurately (e. g., small variations in the terminal temperature difference of the condenser, a heater cut-out or deficiencies of measuring instruments).

Should a control valve which ought to be closed at test load open because of fluctuations in load, or should inadmissibly large fluctuations in test conditions arise during a limited period, this period of the test may subsequently be omitted by mutual agreement of the parties to the test, provided the remaining period fulfils the requirements of 3.8.3. Otherwise the test has to be repeated.

It is recommended that one or more stage pressures and temperatures be observed during the tests, since these may serve as a means of disclosing reasons for any inconsistencies.

3.8.8 *Preliminary calculations*

Preliminary calculations of corrections and test results shall be conducted immediately after the test in order to confirm the validity of the data taken.

3.8.9 *Consistency of tests*

Should serious inconsistencies arise, either during a test or during the computation of results from a series of tests, the test or tests shall be rejected, in whole or in part, unless otherwise agreed.

3.9 *Repetition of acceptance tests*

Should the acceptance test result be unsatisfactory, the supplier shall be given an opportunity to make modifications and to repeat the test at his own expense. A repetition may also be requested by one of the parties to the contract if justified by doubts about the results.

If the supplier, for reasons which are his responsibility, has made modifications following the acceptance test which make it probable that guarantee values will no longer be met within a reasonable margin, the acceptance tests may be repeated at the request of the purchaser.

4. Measuring techniques and measuring instruments

4.1 *General*

4.1.1 *Measuring instruments*

Instruments conforming to any one or more of the following categories are permitted for acceptance tests:

a) Measuring instruments calibrated by an official authority.

b) Measuring instruments, which have been calibrated by comparison with officially calibrated measuring instruments by an official authority.

c) Standardized measuring instruments with known accuracy.

d) Other approved measuring instruments with known accuracy, the use of which has been agreed upon between the parties to the contract.

Table II indicates the available types of instrumentation for acceptance tests.

Typical arrangements in a plant are illustrated in Figures 2 and 3.

If transducers are used, they shall have a suitable range and a demonstrable accuracy equivalent to usual instrumentation.

Measuring instruments and transducers may be equipped with devices to enable them to record on a data logging system.

Data logging systems for automatic correction and recording of measured values may be used. They may also be used for subsequent processing of these values. Their dependability and correctness of operation has to be checked or proven in advance.

4.1.2 *Measuring uncertainty*

The measurement of each quantity entering into the computation of the test result is liable to some degree of error. The test result is subject to a degree of uncertainty depending on the combined effect of all these errors of measurement.

The measuring uncertainty of the instruments and procedures has to be clearly verified by sufficient general information or, if necessary, by special determination. This particularly applies to remote measuring systems and automatic data-collecting systems.

The uncertainty level for each individual measurement shall be chosen in reasonable relationship with the influence of the reading on the test result.

Data are given in Table II for the measuring uncertainty that can be expected for the individual measured variables according to general experience.

The measuring uncertainty of the test results can be calculated from the uncertainties of the individual measurements according to Clause 7.

4.1.3 *Calibration of instruments*

Instruments requiring calibration shall be calibrated before the test. Certificates shall be available to the parties before the test. Any recalibration after the test shall be a matter for agreement between the parties.

4.1.4 *Alternative instrumentation*

By mutual agreement of the parties to the test, advanced instrument systems such as those using electronic devices or mass-flow techniques, may be used as alternatives to the mandatory instrument requirements, provided that the application of such systems has demonstrated accuracy equivalent to that required by these rules.

4.1.5 *Mercury in instrumentation*

Mercury and compounds of mercury react with component materials and cause environmental problems. The very low vapour pressure of mercury presents a serious health hazard if spillage occurs. The greatest care and strict adherence to all applicable regulations concerning mercury is necessary. If mercury is used the following precautions should be taken:

a) Keep instrument valves closed except during test runs.

TABLE II

Acceptable instrumentation for acceptance tests and resulting average uncertainties

Reference	Measured Variable	Measuring Instrument	Accuracy Class	Range	Average Measuring Uncertainty	Remarks
1	Pressure	Deadweight gauge		$P > 2$ bar	±0.3%	
2		Transducer calibrated for test		For all pressures	±0.3%–0.5%	
3		Bourdon gauge calibrated for test	0.3%	$P > 2$ bar	±0.3%–0.6%	
4		Bourdon gauge calibrated	0.6%	$P > 2$ bar	±1%	
5		Mercury manometer		$P < 2$ bar	±1 mm	On length of liquid column
6		Liquid manometer		$P < 2$ bar	±1 mm	
7	Differential pressure	Liquid manometer		$h > 100$ mm	±1 mm	
8		Differential pressure transducer calibrated for test		For all differential pressures	±0.3%–0.5%	
9	Temperature	Thermocouples calibrated		$t \leq 300$ °C	±1 K	
				$t > 300$ °C	±0.5%	
10		Resistance thermometer calibrated		$0 < t \leq 100$ °C	±0.2 K	
				$t > 100$ °C	±0.5%	
11		Mercury-in-glass-thermometer graduated 0.1 °C calibrated	0.10 °C	$t < 100$ °C	±0.1 K	
12	Primary flow	Standardized differential pressure devices			0.75–1.5*	ISO 5167
13		Calibrated differential pressure devices				
14	Cooling water flow	Vane type flow meter		$D > 1000$		IEC 41
15	Electrical output	Two-wattmeter method Instrument transformers calibrated for test Instruments calibrated for test at test load	 0.2% 0.2%		0.1%–0.2%	
16		Three-wattmeter method Instrument transformers calibrated for test Instruments calibrated for test	 0.2% 0.2%		0.1%–0.2%	
17	Current	Ammeter	0.2%			IEC 41
18	Voltage	Voltmeter	0.2%			IEC 41
19	Mechanical output	Torsion dynamometer or thermodynamic method for pump			approximately ±2%	
20	Speed	Manual tachometer		Range of calibration	±1.0%	
21		Manual revolution meter		Range of calibration	±0.5%	
22		Electronic revolution meter		Range of calibration	±0.1%	
23	Ambient pressure	Precision barometer			±0.2 mbar	

* The accuracy of the primary flow measurement is decisive for the measuring uncertainty of the test result according to Table 3 and the pressure differential device shall be chosen accordingly.

b) Install quick closing solenoid valves in instrument pressure sensing lines to close automatically on system upsets.

c) Employ double mercury traps.

d) Locate the primary flow element in the low temperature part of the feed water cycle of nuclear plants where it is remote from the steam supply system.

4.2 *Measurement of power*

4.2.1 *Determination of mechanical turbine output*

The mechanical output of the steam turbine can be determined by one of the following four techniques:

a) By measuring the output at the generator terminals (see 4.2.4) and adding the generator losses.

b) By measuring torque and speed.

Absorption or torsion dynamometers are permissible, provided precautions are taken in their construction and use to ensure accuracy. This includes electric or eddy-current generators, the input of which is measured by the reaction of the stator.

If auxiliary power services of the turbine, such as governor and lubricating oil pumps, are driven from an external energy supply, their power requirements shall be deducted from the turbine output at the coupling in order to determine the net output at the turbine coupling.

c) By establishing an energy balance of the steam turbine.

The output is derived from the algebraic sum of all energy flows passing through the energy balance envelope established around the steam turbine[1].

d) By establishing an energy balance of a driven machine (e.g. compressor, pump).

The output is derived from the algebraic sum of all energy flows passing through the energy balance envelope around the driven machine[1].

4.2.2 *Measurement of boiler feed pump power*

For a complete evaluation and for correction of the test results according to the guarantee definition, it is normally necessary to measure the power consumed by boiler feed pumps. It corresponds to the increase in the enthalpy of the water[2], and possibly the power consumed by the hydraulic couplings and speed changers if the pumps are driven directly from the mainshaft, or by an auxiliary steam turbine driven by a bleed from the main turbine.

If the boiler feed pump is driven by an auxiliary turbine, delivered under the responsibility of the manufacturer of the main turbine, and fed by steam from the main turbine, or is driven direct from the main shaft, it is, according to the guarantee values, normally necessary to measure the power consumed by the pump, corresponding to the increase of the enthalpy of the water and the mechanical losses, and possibly the power consumed by the hydraulic couplings and speed changers.

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[1] Insignificant energy flows, for example, heat dissipated to the environment due to conduction and radiation, can in most cases be estimated with sufficient accuracy and need not be measured.

[2] In many cases mechanical losses and heat losses of the pump can be neglected.

If the use of a dynamometer (see 4.2.1) is not practicable, the pump power is best determined from the rise in specific enthalpy across the pump multiplied by the mass flow.

Since the temperature rise across the pump is small, a high degree of precision is needed in the temperature measurement (e. g. multiple thermocouples connected in series or resistance thermometers in differential connection). In the absence of precise measurements, values of pump efficiency supplied by the manufacturer may be used in conjunction with the measured flow and pressure rise, but unless they are based on tests over the full range of load, they may be open to question.

Sealing water flow and/or injection water flow has to be duly considered in the determination of the mass flow.

The power losses in the pump bearings, the hydraulic coupling and the gear drive are equivalent to the heat picked up in the oil cooling water. Radiation losses are generally negligible.

If more than one pump is in service, it may not be practicable to measure the water flow or the mechanical losses in the bearings, couplings and gear drives individually. However, the total power consumed by the boiler feed pumps in service is all that is needed for an acceptance test.

The power consumed by a pump driven by a rotating electrical machine can be determined from the power consumed by the motor, having regard to the efficiency of the latter according to information supplied by the manufacturer, and from the measurement of the power consumed by the bearings, the hydraulic coupling and the gear drive.

If the auxiliary turbine driving the pump is outside the responsibility of the supplier of the main turbine, it may be sufficient to measure the flow rate and the condition of the bled steam..

4.2.3 *Determination of electrical power of a turbine generator*

The net power of a turbine generator is defined by the following formula:

$$P_g = P_b - P_a \qquad (9)$$

Where an auxiliary of the turbo-generator is driven by an electric motor, the power P_a is the power delivered to the motor. This applies whether the power is derived from the generator terminals downstream of the point where P_b is measured or from a separately generated supply.*

Where an auxiliary of the turbo-generator is driven by other means, for example, a pump or exciter driven by a prime mover, the power P_a is the power input to the coupling.

Where excitation is derived from the generator terminals at a point downstream of the point where the power P_b is measured or from some other source, P_a is the power input to the excitation equipment.

When the turbine generator, condensing and feed water heating plant are guaranteed as a combined unit, the power requirements of the condenser and feed water heating auxiliaries shall be treated in accordance with the terms of the contract.

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* This does not refer to IEC 34-2.

4.2.4 *Measurement of electrical power*

For a three-phase generator with the neutral directly earthed (grounded) or with the four-line system, the unit power shall be measured by the three-wattmeter method.

For a three-phase generator with neutral earthed (grounded) through resistance, reactance, or transformer with resistance, the unit power may be measured by the two-wattmeter method, but preferably by the three-wattmeter method. In all instances, watthour meters may be used in place of wattmeters.

A double measurement of the electrical power, if practicable, including the duplication of voltage and current transformers, has advantages for improvement of accuracy.

4.2.5 *Electrical instrument connections*

The instrument transformers shall be connected into the lines from the generator as near to the generator terminals as practicable and on the generator side of any external connections by which power can enter or leave the generator circuit.

The leads to the meters shall be arranged in such a manner that they will not influence the meter readings by reason of inductance or any other such cause. Inductance may be eliminated by braiding the wires of each pair of conductors for at least 1 m from the position where the instruments are located. It is desirable to check the whole arrangement of meters for stray fields, not only from the instrument leads but also from any other source.

The calibration of the transformers shall be made wherever practicable with the same instruments and wiring impedance as in the tests.

The wiring influence of the voltage circuit shall not cause a significant error in measurement of the power output which will be compared with the guarantee. The choice of cross-sectional area of the wiring shall take into account the length of the wire, the influence of the voltage transformers and the resistance of safety fuses in the circuit. The errors due to wiring resistance (including fuses) shall always be taken into account.

4.2.6 *Electrical instruments*

Singlephase or polyphase portable precision wattmeters, or singlephase or polyphase portable precision watthourmeters shall be used with appropriate voltage and current transformers for measuring electrical output. Any error in each wattmeter or watthourmeter shall not exceed 0.2% of the reading.

Portable ammeters, voltmeters and wattmeters shall be included in the measuring circuits to establish that the generator load conforms to rated conditions during the tests and to measure the current, voltage and power factor.

For the environmental conditions of the instrumentation see 3.2 point *e)* (second paragraph).

The recording time of watthourmeters shall be measured in a manner such that any inaccuracy will not exceed 0.03%. Watthourmeter readings shall be recorded during the test at regular intervals (at least every 5 min).

4.2.7 *Instrument transformers*

Instrument current and potential transformers of appropriate rating and accuracy characteristics, specially intended for test purposes, should be used. Values of ratio and phase-angle corrections for the conditions of loading equivalent to the test instruments and leads used during the test shall be obtained by a recognized calibration procedure to cover the range of test values of current and voltage. Instrument transformers shall be used which have no burden other than that of the test instruments and leads. Otherwise, it has to be confirmed that the permissible burden is not exceeded.

4.3 *Flow measurement*

4.3.1 *Determination of flows to be measured*

In an acceptance test the flows to be measured can be divided into two categories:

Primary flows:	Flows which are in direct relation to the power output and which must be measured with a corresponding degree of precision (see 7.4). To verify the accuracy of measurement of the primary flows, it is recommended that they be measured simultaneously in at least two different locations, and the readings compared.
Secondary flows:	Flows which are necessary for the operation of the plant and which shall be taken into account in the adjustment of the measured values of the primary flows in order to determine the turbine initial steam and reheat steam flows.

4.3.2 *Measurement of primary water flow*

The measurement of primary flow may be made:

a) by means of standardized or calibrated orifices or nozzles;

b) by direct weighing using tanks and suitable scales;

c) by means of calibrated volumetric measuring tanks.

It is seldom practicable or economical to employ weigh tanks or volumetric tanks for the testing of the large units installed in modern power stations. The usual method of determining flows is by means of differential pressure devices.

4.3.2.1 *Differential pressure devices for measuring primary water or steam flow*

Standardized or calibrated differential pressure devices may be used to measure flow.

A choice can be made from among the following recommended devices:

a) Sharp-edged orifice plate (see ISO 5167).

b) Wall tap nozzle (see ISO 5167).

c) Elliptical throat tap nozzle. This device is not covered by ISO 5167 but as it has now become established as an instrument of precision; full details are given in Appendix B.

The diameter ratios of the differential pressure devices shall be selected with due considerations of the resulting measuring uncertainty.

During the passage of the water through the orifices and nozzles either the pressure shall remain not less than 250 kPa above the saturation pressure which corresponds to the measured temperature or the temperature of the flow shall remain not less than 15 K below the saturation temperature corresponding to the lowest absolute pressure in the measuring device.

4.3.2.2 *Calibration of differential pressure devices for water flow*

If a differential pressure device has to be calibrated, this should preferably be done with flow conditions corresponding to the same range of Reynolds numbers that will prevail in the turbine tests. However, the laboratory facilities at present available are not capable of achieving values of Reynolds numbers as great as those associated with test flow conditions for modern large steam turbines. The necessary extrapolation of the discharge coefficient and the resulting difference of the measured and extrapolated value shall be considered in the determination of the measuring uncertainty.

Calibration is to be conducted with the upstream and downstream piping comprising the flow section to be used in the test, including the flow straightener if used.

It should be demonstrated during calibration that a pressure difference device is capable of repeating selected calibration points with sufficient accuracy.

4.3.2.3 *Inspection of differential pressure devices*

It is recommended that shortly before or after the test all differential pressure devices and their pipe sections used for measurement of primary mass flow be inspected for condition (e.g. roughness), size (e.g. sharpness of edge for orifice plates), and general conformity with the standard required. The outcome of this inspection shall be recorded. If differential pressure devices are not inspected, this shall be duly considered in the evaluation of the test according to 5.2.3.1.

These differential pressure devices shall not be installed prior to the cleaning period before commissioning of the plant. They shall either be fitted after cleaning or they shall be bypassed.

In the case of condensing turbines, it is mandatory that practical arrangements for inspection be provided for at least one differential pressure device for a primary mass flow, preferably the condensate mass flow upstream of the deaerator. At least one differential pressure device has to be inspected after the test if this is required by one party.

4.3.3 *Installation and location of differential pressure devices*

The minimum requisite lengths of straight piping upstream and downstream of the orifice or nozzle are influenced by the configuration of the piping before and after the straight sections; for particulars see the ISO standard quoted.

In special circumstances where rotational flow can be expected or when the required straight lengths are not available, application of flow straighteners shall be considered. The incorrect use of flow straighteners may lead to errors. If a flow straightener is used and straight pipe length is not sufficient, the measuring element shall be calibrated with the complete straight piping section upstream and downstream, the flow straightener included.

It is advisable that one of the flow-measuring devices be located in the system at a point where the temperature is less than 150 °C to minimize temperature effects, that is, thermal expansion correction and any distortion of the primary element.

In order to reduce the possibility of thermal distortion of the flow-measuring device, it is desirable that the pipe and the flanges of the flow section be made of corrosion-resistant material having the same coefficient of expansion as the primary element.

When a flow-measuring device for water flow is installed in a vertical run of pipe, a correction may be necessary to account for differences in altitude between the two pressure-tapping points and differences in density between the water flowing through the flow-measuring device and the water in the pressure-tap lines.

To minimize the difficulty of obtaining steady flow, the flow-measuring device should not be located at a pump discharge. Advantage should be taken of the damping effect of any existing heat exchangers and long lengths of pipe in the cycle. The location of the flow-measuring device should also be such that the effects of recirculating and bypassing flows are eliminated. If this is not practicable, extraneous flows shall be measured with sufficient accuracy.

If the feedwater heating cycle includes a deaerator, it is recommended that the condensate flow entering it be measured. This eliminates any possibility of heater leakage recirculating through the flow-measuring device. If the cycle has no deaerator, it is recommended that the flow be measured after it has passed through the low-pressure heaters and before it enters the boiler feed pump. If the drains from the high-pressure heaters join the main condensate flow upstream of the flow-measuring device, it will be necessary to measure the total drain flow from the high-pressure heaters and calculate the extraction steam by heat balance around the heaters to determine the high-pressure heater leakage.

For turbines using wet steam and heaters with pumped-ahead drains, a suitable double flow measurement may be required to determine if any of the heaters leak. The location of the low pressure flow measurement will depend on the cycle arrangement. Heater leakage can also be measured by using the tracer method.

4.3.4 *Differential pressure measurements*

The measurement of the differential pressure necessitates particular care. Manometer systems shall be installed with the precautions listed below.

Measuring accuracy for primary flows can be increased by the use of two independent manometer systems.

a) Connecting piping used between the pressure taps and the manometers shall be not less than 6 mm inside diameter tubing of material which will minimize resistance damping inside the piping. This tubing should run horizontally for 1 m from the flow-measuring device and then slope down continuously without loops to the manometers. The connecting piping shall be shown to be tight by a pressure test.

b) The length of piping between the flow-measuring device and manometers should not exceed 7.5 m if practicable. It shall also be uninsulated.

c) Precautions shall be taken when running the manometer piping to ensure that any temperature difference between the fluid in the two lines connecting the primary element and each manometer remains negligible. It is recommended that the piping be bundled and run in such a manner as to minimize heat transfer to these pipes from an external source.

d) The manometer piping shall be well cleaned, preferably by flushing, before the manometer is connected. The manometer connections shall include the necessary means for shutting off manometer piping or venting at any time during the test. Sufficient time should be allowed for the water legs in the connecting piping to reach temperature equilibrium.

e) Zero-displacement solenoid-operated valves may be installed in each tube close to the primary element to eliminate mercury column movement during reading. These valves are to be closed for reading at prescribed intervals without regard to position of the mercury column. Other means of obtaining instantaneous readings may be employed if they do not introduce errors in the reading. Solenoid valves should not be used with transducers.

f) The manometers should be located at a lower elevation than the primary-flow element. However, when this is impracticable, special precautions shall be taken to ensure proper venting of the system. Suitable venting pots shall be installed above the manometers with valves for venting. Also, a temperature seal (loop in piping) shall be installed between the primary element and the manometers.

g) The manometers should be precision-type and should be read with the aid of antiparallax readers or other suitable means. The mercury shall be instrument-grade having less than one part per million of non-volatile residue. The manometer shall be scrupulously clean before the mercury is introduced.

h) When pressure transducers are used, they shall be calibrated before the tests. Each calibration shall be done with rising and falling pressure. The long-term stability of the transducers shall be carefully recorded. The tests are to be calculated with the mean calibrated values.

i) For the measurement of steam flow, venting pots should be installed at the same elevation as the primary element pressure taps, and at a suitable distance to ensure proper condensation. The water levels shall be at the same elevation, otherwise differences are to be determined. The connecting piping to the water pots should have a sufficiently large diameter to avoid the formation of water plugs.

The connecting lines to the manometer shall be installed with a continuous downward slope. After venting, time should be allowed for the water legs to form and cool. Special attention has to be paid to the proper formation of water legs at sub-atmospheric pressure.

4.3.5 *Water flow fluctuation*

Flow measurements shall not be attempted unless the flow is steady or varies only slightly with time. Variations in the flow shall be suppressed before the beginning of a test by very careful adjustment of flow and level controls or by introducing a combination of capacitance (such as bypassing flow around the pump) and resistance (such as throttling the pump discharge) in the line between the pulsation sources and the flow-measuring device. Damping devices on the manometers do not eliminate errors due to pulsations and, therefore, shall not be used. If the pulsations remain excessive after every effort has been made to suppress them, mutual agreement is required before the test can be started. Electrical locking of a digital reading is also advisable.

4.3.6 *Secondary flow measurements*

In addition to the primary flow measurements (see above), many secondary flow measurements are required. Because of differences in plant layout and the alternative positions of flow-measuring devices, it is not possible to specify the accuracy required in each separate measurement. The parties to the test should decide what secondary flow measurements shall be made for the type of turbine installation concerned and then arrange for each measuring device to have an accuracy such that the combined effect of all errors in the secondary flow measurements is in appropriate proportion to the error in the test result. Consequently, if standardized measuring devices are used, calibration may not be necessary.

If the secondary flow-measurement is of steam, the steam shall remain superheated in passing through the flow-measuring device. Measurement shall not be attempted if the amount of superheat is less than 15 K at the smallest cross-section of the nozzle or orifice.

4.3.6.1 *Extraction steam flows to feed heaters*

If the extraction steam is superheated the extraction flow can be determined by heat balance calculation. The accuracy of the result falls off as the temperature rise across the heater diminishes.

A direct measurement by suitable pressure differential devices is possible, if the necessary measuring accuracy can be achieved.

Wet steam extraction flows can be determined from heater drain flow measurements. This can be accomplished by using a standardized or, if necessary, a calibrated flow measuring device with suitable installation conditions.

The heater drain flows can be measured using differential pressure devices except for the lowest pressure heaters with cascaded drains where very little pressure drop is available. In these cases, Venturi tubes or other low head loss primary elements of the required accuracy should be used. The differential pressure should preferably be measured with a transducer. The connecting piping between the transducer and the differential pressure device should be as short as practicable to minimize damping errors due to unsteady flow and care shall be taken to eliminate air bubbles in this piping.

If, for these reasons, transducers are mounted in areas not accessible during operation of the plant, suitable means for a remote operated calibration are required. The calibrating reference should be a secondary standard.

Heater drain flows are often very unsteady; therefore, the transducer output should be observed every 20 s.

In sizing the primary elements to avoid cavitation, the best compromise between Reynolds number, pressure loss, diameter ratio, and deflection should be made without reducing the critical cavitation coefficient "K" below 0.2, where:

$$K = \frac{p_{\text{throat}} - p_{\text{sat}}}{\frac{\varrho}{2} \cdot W^2_{\text{throat}}} \qquad (10)$$

The cavitation problem may be reduced by providing a loop seal or increasing the length of the loop seal to put more head on the meter so that the critical cavitation coefficient will be greater than 0.2.

If the heater drain flow is subcooled the cavitation problem is also reduced. The extraction wet steam flow can be determined by a heat balance calculation, if the enthalpy of the extraction steam has been determined by the tracer method (see 4.7.2).

4.3.6.2 *HP heater drains*

When the main flow meters are at the deaerator outlet, the flow from the HP heater drains shall be metered by a device capable of sufficient accuracy, unless the HP heaters have been checked for leakage.

4.3.6.3 *Moisture separator and reheater drains*

It may not be possible to install a flow-measuring device because the available head is less than that required to avoid cavitation and also for reasons of plant design or operation. In these circumstances, the tracer technique can be used to measure the flow with a sufficient accuracy. In case of measurement with a pressure difference device the recommendation of 4.3.6.1 shall be followed.

4.3.6.4 *Boiler feed pump turbine steam supply*

Steam consumption of a boiler feed pump turbine is preferably to be measured as condensate if a separate condenser is installed.

In a reheat cycle having a boiler feed pump driven by a turbine supplied with steam from the main turbine at a point up-stream of the reheater, the steam supplied shall be measured to enable the flow through the reheater to be determined.

4.3.6.5 *Turbine gland leakages*

The gland leakage flows of reheating condensing turbines which do not pass through the reheater but are returned to the system beyond the reheater, have to be determined separately, preferably by measurement, as these flows shall be considered when determining the heat supplied by the reheater.

If any glands are provided with leak-offs to atmosphere or to a point that is extraneous to the turbine system, this flow should be measured unless agreed to be negligible.

4.3.6.6 *Desuperheating spray water flow*

When spray water from the feed-heating system is used for regulating the reheat steam temperature, the rate of flow of such water shall be measured. Also, if the initial steam temperature is similarly regulated, the rate of flow of spray water to the superheater shall be measured, unless the water is drawn from a position downstream of the final feed heater and the last feed water flow measuring device.

4.3.6.7 *Boiler feed pump gland sealing and balance water flow*

The amounts of water supplied to the feed pump glands for sealing and/or cooling purposes and the amounts of leak-off water returning to various parts of the system are liable to be considerable, which may be direct additions to, or subtractions from, the main flow measurement. The number of boiler feed pump water flows to be measured will depend on the location of the main metering device. Calibrated or standardized measuring devices are required and it should be ascertained at the time of the test that they are in good condition.

4.3.6.8 *Stored water changes*

Changes in stored water quantities within the test cycle shall be taken into account in the assessment of the condensate/feedwater flows through the system.

Such stored water quantity changes will include those in condenser hot-wells, boiler drums, deaerator storage tanks, feed heater bodies and any storage or drain tanks that cannot be isolated from the system.

Water level changes in all storage vessels should be measured by temporary scales rigidly fixed close to the sight glasses of the permanently installed level gauges, or, alternatively, by transducers used in conjunction with test data-logging equipment.

If the temperature of the water in a vessel is appreciably different from the ambient temperature and an external gauge glass is employed, for example, as in the case of a deaerator storage tank, the density of water in the gauge glass should be used in the conversion of water level change to mass change in the vessel.

Gauge glasses which are connected to vessels containing hot water should not be blown down within a period of approximately half an hour before taking a reading in order to avoid a false level indication due to a change of temperature in the water column.

Since time is a critical factor in the measurement of the water level changes, the test readings shall be closely synchronized with the signals for commencement and termination of a test.

4.3.6.9 *Leakage determination*

It will be necessary to use calculated values for internal pump leakages, shaft packing, valve stem leakages, internal turbine leakages and other leakages when it is not practicable to measure these flows.

4.3.7 *Occasional secondary flows*

Further secondary flows, which occur relatively rarely, or need not be measured, include the following:

4.3.7.1 *Ejector steam*

Steam-jet air ejector steam flow can be calculated from the measured pressure and temperature of the steam supply and the known cross-sectional area of the jets. When the steam supply is wet, it may be preferable to use the design flow rates given by the manufacturer.

The amount of steam removed from the condenser by the air removal means is generally negligible. Should it be measured, the method of measurement shall be agreed upon by the parties to the test.

4.3.7.2 *Make-up flow*

Make-up flow to the system, if it cannot be avoided, shall be determined.

4.3.7.3 *Water seals*

Sealing water is used in connection with hydraulic glands or atmospheric exhaust valves, condensate pump glands, etc. As the seals have to be maintained, the flow of water used shall be measured or estimated and an appropriate allowance made, if it influences the result of the test. It shall be ensured that there is no variable storage in the sealing system and no possibility of leakage of sealing water other than into the condensate system. If external leakage of condensate water from seals cannot be avoided, its flow shall be determined and added to the condensate flow.

4.3.7.4 *Auxiliary exhaust steam*

Any auxiliary exhaust steam that normally enters the condenser shall either be diverted elsewhere during the period of the test, or measured. In deciding upon the best position to install the measuring device, particular attention should be paid to the net head loss and the possibility of flashing in the throat.

4.3.8 *Density of water and steam*

The density of water which is required for calculation of mass flow rate is calculated from an accurately measured temperature and reasonably approximate pressure. The temperature shall be measured with a precision-calibrated instrument. If an extra instrument is used, it should be located at least 10 pipe diameters downstream of the primary element. For primary water flow, it is also permissible to use the average of the discharge temperature of the upstream heater and the inlet temperature of the downstream heater, providing no extraneous flow enters between them and the temperature is taken at least 10 diameters downstream from the heater outlet to ensure adequate mixing.

The density of steam which is required for calculation of mass flow rate is calculated from measurements of pressures, using a precision gauge, and temperature using a precision instrument or resistance thermometer. The reference planes for the density determination are to be taken according to the calibration procedure or the standard of the flow measuring device.

4.3.9 *Determination of cooling water flow of condenser*

This flow is normally only required if condenser performance is included in the turbine generator performance guarantee.

In many cases a direct measurement is not possible or practicable because of technical difficulties. In these instances, the cooling water flow can be determined conveniently by a heat balance calculation.

The quantity of the condenser cooling may be calculated. If measurement is feasible, one of the following methods may be used:

a) standardized nozzles or orifices in the pipe line;

b) Venturi tubes, or their equivalent in the pipe line;

c) current meters;

d) Pitot tubes, provided it is agreed that the differential head is sufficient to give the necessary accuracy;

e) weir-notch method.

Further measuring methods necessitating less installation effort are available and may be used with sufficient knowledge and care:

f) dilution methods using chemical or radioactive tracers;

g) ultrasonic techniques.

4.4 *Pressure measurement (excluding condensing turbine exhaust pressure)*

4.4.1 *Pressures to be measured*

The initial pressure of steam supplied shall be measured in the steam line on the steam-generator side of, and as close as practicable to, the turbine stop valve and upstream of the strainer if this has been furnished by the manufacturer under the turbine contract. The initial steam pressure shall be measured in the pipe downstream from the parts which have not been furnished under the turbine contract, unless the turbine contract or specification states otherwise.

The steam strainer shall be known to be clean. If there is a doubt about its cleanliness on the part of either of the parties to the test, it shall be examined prior to the test and cleaned if necessary.

Pressures should also be measured at the inlets to the HP, IP and LP cylinders of the main turbine, at the inlet and outlet of the boiler feed pump turbine (if integrated with the feed-heating system) and at both ends of bled steam lines; also at the suction and discharge of all pumps in the condensate and feed system.

Wherever practicable, pressure tappings shall be situated in straight runs of piping, remote from any flow disturbances.

The pressures measured during steam turbine tests shall be static pressures.

4.4.2 *Instruments*

Deadweight gauges, Bourdon tube type test gauges or mercury manometers shall be used. All of these instruments can be replaced by transducers of suitable measuring range and equivalent accuracy (see 4.4.2.5).

Pulsations of pressure shown shall not be damped by throttling on the gauge valve or by the use of commercial gauge dampers. A volume chamber may be employed.

4.4.2.1 *Measurement of pressure above 250 kPa (2.5 bar)*

Bourdon gauges or preferably deadweight gauges shall be used for measuring the pressures above 250 kPa. These gauges shall be mounted in locations as free as practicable from vibration, dirt and excessive variations in ambient temperatures.

For the measurement of pressures where the highest degree of accuracy is not essential, calibrated test Bourdon gauges may be used.

4.4.2.2 *Measurements of pressure below 250 kPa (2.5 bar), but above atmospheric pressure*

Where the level of pressure permits their use, liquid manometers are recommended.

For single-leg manometers the bore should be uniform and preferably not less than 9 mm to avoid excessive capillary effects.

4.4.2.3 *Measurement of pressures below atmospheric pressure*

Mercury manometers (see 4.1.5) shall be used for measuring pressures below atmospheric.

The tubing for mercury manometers shall be high-grade lead-free glass, preferably not less than 10 mm bore in the region where readings will be taken.

Where small pressure differences must require to be measured with high accuracy, silicone oil may advantageously be used as the manometer fluid in place of mercury.

4.4.2.4 *Liquids for manometers*

The liquid used shall be suitable for the application and of known density.

4.4.2.5 *Transducers*

Accurate pressure measurements are possible with transducers, provided that their use and care are well understood and that they are properly maintained and installed. Regardless of the application, each transducer should be calibrated before the tests.

Each transducer should be located in a position which is free from vibration and dirt, and where there are not likely to be large changes in ambient temperature such as may be caused by an outside door.

If a transducer is sensitive to changes in environment, such as temperature, the system should be given sufficient time (e. g. 2 h for a quartz Bourdon type transducer) to stabilize before readings are taken. A zero reading shall be taken before and after each test run.

4.4.3 *Pressure tapping holes and connecting lines*

Tapping holes should be at right angles to the inner surface of the pipe. The inner rim of the hole should be free from burrs, having its edge sharp and square. For a length of at least twice its diameter, the hole should be straight and of uniform bore. The bore of the tapping holes should be, from 6 mm for higher pressure to 12 mm for lower pressures.

To prevent errors due to an unknown head of water accumulating in the line connecting the tapping hole to the pressure measuring device, the installation should always be arranged to have the connection lines either completely full of water or completely empty.

4.4.3.1 *For pressures above 250 kPa (2.5 bar)*

For higher pressures in the range above that of liquid manometers, it shall be ensured that the connecting lines are full of water. The gauge can be above, or preferably below, the tapping point. The tapping hole should preferably have a 6 mm diameter.

4.4.3.2 *For pressures below 250 kPa (2.5 bar), but above atmospheric pressure*

The gauge should be situated below the tapping. Although small bore connecting lines may be considered, they are prone to becoming blocked, so connecting lines not less than 12 mm in diameter are recommended.

For pressures above atmospheric but low enough to be measured with liquid manometers, the connecting lines may be arranged to be either full or empty. The chances of incurring an error due to unknown heads of water in the connecting lines are high in this pressure range and a condensing vessel between the tapping and the pressure measuring device should be provided.

4.4.3.3 *For pressures below atmospheric pressure*

For pressures below atmospheric the connecting lines should be free from water. The tapping hole should preferably be 12 mm in diameter, the pressure measuring device should be above the tapping and there should be a continuous slope back to the tapping.

If the tapping hole is only 6 mm in diameter, then the connecting line (or the greater part of it) should be thick-walled, non-metallic tubing to minimize condensation and lessen the possibility of water collecting in the line. The connecting lines should preferably be 12 mm in diameter. If small bore connecting lines are used then some form of air purging between readings should be provided.

4.4.4 *Shut-off valves*

A suitable shut-off valve should be installed at each pressure tapping point, and for high pressures there should be a second valve at the gauge end of the connecting line.

4.4.5 *Calibration of pressure measuring devices*

The calibration accuracy of test pressure measuring devices should be within ±0.2% of the pressure being measured, except that for measurements which are not of primary importance this accuracy should be within ±0.5%. The parties to the acceptance test should agree, at a pre-test meeting, which pressures, if any, may be measured to the lower standard of accuracy.

All the test pressure gauges and transducers (but not liquid manometers) should be calibrated against a dead-weight tester or standard manometer before the acceptance tests. Bourdon tube type instruments should also be calibrated immediately after the tests. If there is no reason for preferring one calibration to the other a mean of the two calibrations should be used.

When transducers with data loggers are used the calibration should be made by comparing the true pressure as shown by the dead-weight tester against the typed print-out of the data logger. If the data logger also has a tape output for feeding into a computer it should be demonstrated that the typed output and the tape output are identical.

If differential pressure Bourdon gauges or transducers are used for measuring reheat pressure drops or bled steam line pressure drops, two dead-weight testers should be used to apply the working pressures to the differential pressure gauge for calibration.

Liquid manometers need not be calibrated if the parties using them produce evidence to establish the accuracy of the manometer scales and the purity or density of the manometer fluid. In the case of single leg manometers, evidence of the uniformity of the bore of the tube and the cross-sectional areas of the tube and reservoir should also be available.

If a test manometer equipped to record on a data logger is capable of being used manually, it should be ascertained that the typed print-out agrees with the manual observations. If a test manometer used in conjunction with a data logger cannot be read manually then it should be treated as a transducer and calibrated against a manometer of proven accuracy.

A transmitter may also be calibrated by applying a measured column of low density fluid directly to the low pressure side of the sensitive element and applying simultaneously a high density fluid to the high pressure side of the sensitive element. These two columns then provide the differential pressure. The required working pressure from a pressure source is now simultaneously applied to both columns taking the precaution of ensuring that the pressure medium used does not mix with either the low density or high density fluid. The lengths of the columns are observed and read off by means of high pressure sight glasses.

4.4.6 *Atmospheric pressure*

The atmospheric pressure to which liquid manometers or columns are referred shall, wherever practicable, be determined by means of a precision barometer. The barometer shall be of the mercury-in-glass type, certified by a recognized national authority, with the bore of the glass tube not less than 6 mm. Alternatively, an aneroid or other type of barometer may be used, provided that its accuracy and suitability are certified by a recognized authority.

The barometer should be located in the same room as that of the manometer and as near as practicable at the same height.

If a barometer is not available, the atmospheric pressure shall be determined by the reading taken at the time of the test from a local recognized weather bureau corrected for the difference of elevation between the weather bureau and the turbine.

4.4.7 *Correction of readings*

The readings should be averaged over the test period and then corrected as follows:

All pressure-measuring devices with a head of water in the connecting line should be corrected by adding the pressure equivalent of the head of water when the gauge is above the tapping and subtracting it when the gauge is below the tapping.

A calibration correction shall be applied to those pressure-measuring devices which have been calibrated.

The absolute pressures measured by means of liquid manometers and mercury-in-glass barometers shall be calculated in accordance with ISO 31/111 taking into account:

a) the average of readings;
b) the correction of the length of the scale on account of temperature;
c) capillarity corrections for single leg manometers, unless they have tubes of not less than 12 mm diameter;
d) density of the liquid (taking into account the liquid in the opposite leg);
e) local acceleration due to gravity;
f) for liquid manometers only, the ambient pressure duly corrected;
g) difference in elevation and gravity, of tapping point and instrument, with density of the fluid.

4.4.7.1 *Corrections for the water head*

To obtain the correct pressure at the tapping point, the pressure equivalent of the head of water between the steam pipe tapping and the gauge centre line shall be added (when the gauge is above the tapping) or subtracted (when the gauge is below the tapping) from the gauge reading:

$$\Delta p = H \varrho g \qquad (11)$$

where:

H = vertical distance between the tapping point and the centre of the pressure gauges
p = density of water at the ambient temperature
g = local acceleration due to gravity

4.4.7.2 *Barometer reading corrections*

Mercury-in-glass barometers shall first be corrected at 273 K with appropriate allowance for the reference temperature of the scale. The reference temperature for the scale in metric countries is usually 273 K.

Barometers shall secondly need to be corrected for capillary depression of the mercury. The scale of a mercury-in-glass barometer may have been set to correct for this, in which case no further capillarity correction shall be applied.

Barometers are thirdly to be corrected for difference in elevation, if any, between a barometer and any of the mercury pressure reading devices that are referred to it.

A fourth correction shall be made for gravity at the location of the test. Values of the various correction factors may be found in several recognized tables of standards such as those of the World Meteorological Organization, the Tables of the National Weather Bureau, the Smithsonian Tables, etc.

4.5 *Condensing turbine exhaust pressure measurement*

4.5.1 *General*

The test instrumentation should give the mean static pressure at the exhaust from each individual LP casing. It should normally consist of separate sets of pressure sensing holes, preferably connected individually or consecutively by use of suitable switching devices.

The following rules are intended to apply to all condenser arrangements. It is realised, however, that it may be easier to comply with them in respect of underslung condensers than in the case of side-mounted condensers, particularly of the integral type. Should it be found virtually impossible to comply with the sub-clauses below, then the parties concerned should mutually agree on some alternative means of making the measurements, being guided as far as possible by the concepts herein.

4.5.2 *Plane of measurement*

Unless specified otherwise in the contract documents, the condenser inlet shall be taken as the measuring plane for both turbine and condenser purposes. Modern condensers are of four general types: transverse underslung, axial underslung, separate side-mounted, integral side-mounted. For the first type, the condenser inlet generally is identified with the turbine exhaust flange. For the other three types of condenser it is more difficult to identify the condenser inlet. In such cases the condenser inlet should be placed in one or more planes as close as practicable to the condenser tube nests.

4.5.3 *Pressure taps*

Considerable variations in static pressure usually exist across the exhaust so that it is necessary to make provision for measuring the mean at the condenser inlet by providing numerous pressure sensing holes. Special locations of demonstrable accuracy may be used when agreed upon by the parties to the test, but in no case shall there be fewer than 2 per exhaust annulus. When test results are not available to locate sensing holes one device should be employed for each 1.5 m^2.

For small exhaust conduits requiring not more than four gauges where the walls are straight in the direction of the flow and where the flow is likely to be uniform, flush wall tappings may be used.

The position of each wall tapping should be chosen so that it is, as far as practicable, not influenced by bends, bellows pieces, splitters, gusset plates, staybars or similar disturbances to flow.

The holes should be normally 10 mm in diameter and never less than 8 mm in diameter. They should be drilled normal to the wall, all burrs shall be removed and a wide area round the hole should be dressed clean.

Gauge connections through the walls of the exhaust conduit or through ribs traversing the steam space shall be normal to and flush with the surface of the wall. The hole at the open end should be 12 mm in diameter and uniformly rounded to a radius not exceeding 0.8 mm.

If the configuration of the exhaust is such that wall measurements are unrepresentative, internal devices, including special pressure taps with guide plates, baskets or equivalent devices, should be used and they may also be used if a wall is inaccessible.

Any internal device shall have been shown to be insensitive to steam incidence, and be mutually acceptable to the parties concerned.

4.5.4 *Manifolds*

Pressure averaging manifolds should not be used.

4.5.5 *Connecting lines*

Special precautions shall be taken in making the connections to the exhaust conduit. Each mercury manometer or column shall be mounted as near as practicable to the corresponding pressure connection in the exhaust conduit or casing, but in a position where it will be free from excessive vibration and where the observer can take readings with convenience and accuracy.

The test gauges should be at a higher level than the sensing holes, so that the measuring system can be self-draining back to the condenser. Alternatively, special arrangements for adequate draining may be made. If it should prove difficult to devise a self-draining system, then the parties interested in the tests should mutually agree on a means of adequately draining or purging the system by air or nitrogen flushing. Other requirements for low-pressure connecting lines appear in 4.4.3.

4.5.6 *Instruments*

Any reliable instrument for measuring the exhaust pressure may be used, including open-ended liquid manometers and a barometer, closed liquid manometers (absolute pressure gauges) or pressure transducers. The reading accuracy should be 35 Pa.

An accurate determination of site barometric pressure is required (see 4.4.6).

4.5.7 *Tightness of measuring system*

The tightness of the measuring system should be checked. When the valves adjacent to the sensing holes are closed under specified vacuum, the column reading, if mercury is being used, should not fall at a rate greater than 6 mm in 5 min. Rubber diaphragm type valves are more effective than cocks in keeping the system tight.

4.5.8 *Calibration*

In accordance with the requirements of the fifth paragraph of 4.4.5, liquid column vacuum gauges need not be calibrated if the parties using them produce evidence to establish the accuracy of the manometric scales and can show that the purity or density of the liquid is satisfactory. Corrections for meniscus and capillarity are unnecessary if mercury manometers have 12.0 mm bore tubes or larger.

If pressure transducers are used they should be calibrated before and after the test against a liquid manometer or vacuum dead weight tester.

If a liquid column gauge is equipped to record on a data logger and can also be read manually as a liquid column, then the requirements of the sixth paragraph of 4.4.5 apply.

4.5.9 *Correction of readings*

The readings should be averaged over the test period and then corrected in accordance with 4.4.7.

Liquid column readings should be corrected in accordance with 4.4.7.

4.6 *Temperature measurement*

4.6.1 *Points of temperature measurement*

Temperature measurements shall be made at points as close as practicable to the points at which the corresponding pressures are measured for enthalpy determinations. Temperatures, the values of which influence the results of the test, shall be taken at two different points close together and the mean of the two readings shall be taken as the temperature of the fluid.

Should it be suspected that there are distributions of flow having different temperatures within a pipe, and the weighted average temperature influences the results of the test, the pipe is to be explored across its diameter by temperature measurement means and an average determined by a means to be agreed by the parties to the test.

4.6.2 *Instruments*

The preferred instruments for measuring relatively high temperatures are:

a) calibrated electrical resistance thermometer with calibrated precision bridge or digital voltmeter;

b) calibrated high-grade thermocouple and precision bridge or digital voltmeter. Continuous leads to the cold junction are recommended for high measuring accuracy.

The thermocouples and resistance thermometers, and their potentiometers, bridges and galvanometers or digital voltmeters should be calibrated before the test or at regular intervals where sufficient stability has been shown. Otherwise the increased uncertainty of measurement has to be taken into account in accordance with 7.2.2.

Recalibration after the tests will generally not be necessary if individual temperature measurements are sufficiently cross-checked.

All this equipment shall be carefully handled and maintained. Its condition shall be examined periodically.

Additional mercury-in-glass thermometers are recommended for use when the temperature to be measured is less than 373 K and the locations are such that readings may be taken without difficulty.

Mercury-in-glass thermometers used to measure temperatures which influence the results of the test shall be of the solid-stem precision type with etched scale plate graduated with scales suitable for the measurements to be made. Commercial or industrial metal-encased thermometers shall not be used.

4.6.3 *Main temperature measurements*

The main temperatures are those which have a direct influence on the acceptance test result, such as initial steam, hot and cold reheat steam, final feed water temperature, and cooling water temperatures.

Other temperatures, such as feed water pump inlet temperatures and discharge and deaerator condensate inlet temperatures, may also be main temperatures according to the guarantee formulation, and to the method of determining the primary flow.

If the fluid flows associated with the main temperature measurements are conveyed in multiple pipes, each main temperature should be taken as the arithmetic mean of the individual temperatures unless the parties interested in the test agree that some other flow weighting is appropriate.

Two thermometer wells should be provided in each pipe to permit the temperature measurement to be independently duplicated. The temperature in each pipe should be taken as the arithmetic average of the two measurements unless the parties to the test agree otherwise.

The final feed temperature measurements should be taken after the junctions of any heater by-passes, and sufficiently far downstream to ensure that adequate mixing has occurred.

4.6.4 *Feed train temperature measurements (including bled steam)*

These temperatures are normally only required when the feed water heating plant (if installed) is not provided by the turbine contractor, although it may be of advantage to measure some or all of them.

With the exception of main temperature measurements, the feed train temperature measurements need not be duplicated. Sufficient inlet and outlet temperatures on each heater should be measured to enable a heat balance to be calculated.

In many instances, the inlet feed temperature of one heater would be the same as the outlet feed temperature of the preceding heater* and, in these instances, one temperature measurement to serve both heaters would be sufficient. If both the feed inlet temperature and the preceding feed outlet temperature are measured, then the mean of the two should be taken to obviate discrepancies in the heat balance. Where a return connection exists between the two points, measurements should be made upstream and downstream of the junction.

Steam temperatures should be measured at each end of the bled steam pipes: near the turbine connection upstream of the non-return valve and near the feedwater inlet connection.

If mixing occurs in a bled steam pipe, the temperature of the mixture should be taken sufficiently far downstream of the junction to ensure that adequate mixing has occurred.

Thermowells should be located in areas which assure good mixing and have minimum possibility of stratification.

4.6.5 *Condenser cooling water temperature measurement*

These temperatures are normally only required if condenser performance is included in the turbine generator performance guarantee.

a) Inlet temperature
Inlet temperature is generally constant across the section of the pipe, and unless there is reason to suspect that stratification is occurring, one temperature measurement in each

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* The temperature is taken at least 10 diameters downstream from the heater outlet to ensure adequate mixing.

inlet pipe will suffice. Thermometer pockets may be used or, alternatively, a continuous flow of sample water may be drawn off through a chamber into which is inserted a direct contact thermometer. If either party suspects stratification, the measurement shall be by one or both of the methods used for outlet temperature as described in the following paragraph.

b) Outlet temperature

Temperature stratification occurs in the cooling water discharge boxes. To permit mixing to occur, the measuring positions should be several diameters downstream. Temperature sampling probes should be used at each condenser outlet, on at least two diameters. The probes may be either perforated sampling tubes through which water can be drawn off to a mixing chamber or they may be thermopiles to obtain a mean temperature in situ. There shall be not less than one sampling hole or thermocouple for each 0.2 m^2 of pipe area and they should be situated at centres of equal areas. There should be not less than four sampling holes or thermocouples for each outlet pipe and they should be situated in such a way that they are representative for the cross-section of the pipe. If practicable, the temperature can be measured after mixing in the outlet water channel with due consideration for temperature loss.

4.6.6 *Accuracy of temperature measuring equipment*

The temperature measuring equipment shall be of such accuracy that the individual maximum errors do not prevent the desired level of test accuracy from being achieved.

4.6.7 *Thermometer wells*

The material of a thermometer well shall be suitable for the temperature to be measured. Tubes and wells shall be as thin as practicable, consistent with safe stress, the inner diameters being as small as practicable. It is important that the wells be clean and free from corrosion or oxide. Wells for high or main temperature measurements may be provided with external fins for good heat absorption. With high pressures and high temperatures, it is advisable to weld the thermometer wells to the pipe.

Thermometer wells shall preferably be dry, especially for the higher temperature measurements but shall be carefully covered and sealed with material to reduce air circulation or loss of heat.

If temperature rise across the feed pump has to be measured, the thermometer pockets at inlet and discharge shall be of duplicate type and material. The pocket in the discharge piping shall be located sufficiently downstream of the pump to permit a good mixing of the flow.

4.6.8 *Precautions to be observed in the measurement of temperature*

The following precautions shall be observed in the measurement of the temperature:

a) Heat that is transmitted by conduction or radiation to or from the temperature measuring means other than from the medium being measured shall be reduced to a minimum.

b) The immediate vicinity of the point of insertion and the projecting parts of the well and its support shall be thermally insulated.

c) In pipes of less than 75 mm internal diameter, the thermometer shall be arranged axially in the pipe by inserting it in an elbow or tee. At any location where an elbow or tee is not available, the piping shall be modified accordingly.

In pipes having an internal diameter greater than or equal to 75 mm, the sensitive element shall be located between the centre and a point 25 mm radially beyond, except that in the case of very large pipes and special multi-point measuring devices, no well need exceed a length of 150 mm inside a pipe.

d) In measuring the temperature of fluids the heat-receiving part of the apparatus shall not be in a dead space.

e) In taking readings, a mercury-in-glass thermometer shall not be raised in the thermometer well further than is necessary to render visible the top of the mercury thread. Other types of instrument shall not be moved while readings are being taken.

f) Each complete temperature measuring installation shall be set up in place in full working condition at the temperature conditions that will prevail during the test, for not less than 2 h before a test is commenced.

4.7 *Steam quality measurement*

4.7.1 *General*

In power stations employing certain types of nuclear reactor, the steam supplied to the turbines is at saturation temperature and may contain a small percentage of moisture. It is therefore necessary to ascertain the moisture content in order to determine the enthalpy of the initial steam.

It may also be required to determine the moisture content of the steam after partial expansion in the turbine, for example, at a point where extracted steam is taken to a feedwater heater or a water-separator or reheater. Where it is practicable to condense the extracted steam and measure the condensate, the enthalpy of the steam can be determined by a heat balance calculation.

The usual methods of determining the moisture content of steam are by means of

- a throttling calorimeter,
- an electrical heating calorimeter.

The former is only applicable when the quality and the pressure of the steam are sufficient to produce a measurable degree of superheat in the calorimeter. The latter is not subject to these limitations. Both methods, however, are liable to give misleading results because of the probable lack of homogeneity and the difficulty of obtaining a correct sample of the local steam-water mixture. Consequently, there is uncertainty as to whether the steam passing to the calorimeter is representative of the main steam flow.

Recently, more precise methods employing radioactive or non-radioactive tracers have been introduced and established.

4.7.2 *Tracer technique*

The tracer technique employing the dilution method is an accurate method for determining either the water phase or the moisture fraction of a two-phase vapour-water flow. The dilution method is based on the measurement of a tracer concentration in a water sample. Dilution can be accomplished in either of two ways, applicable to determining both throttle and extraction enthalpy.

Condensing method: It is assumed that the tracer concentration in the moisture carried by the steam is the same as that in the water in the boiler. After condensation of the vapour and measurement of the mass flow and of the tracer concentration in the boiler and in the condensate, a mass balance calculation enables the amount of water transported in the steam from the boiler to be determined.

Constant rate injection method: A solution of tracer of known strength is injected at a known rate into the steam flow. After mixing, a sample of the water is drawn off and the tracer concentration measured. A balance calculation enables the moisture content of the steam upstream of the injection point to be determined. Precautions shall be taken to ensure that the mixture is homogeneous and that no steam is withdrawn with the water.

Methods other than these may be used upon agreement.

4.7.3 *Condensing method*

An appropriate tracer, dissolved in the water phase of wet steam at a concentration C_{wat} will be diluted by condensation of vapour. After the steam is totally condensed, the tracer concentration in the condensate will be C_{cond}. The concentrations are related by the balance:

$$C_{wat} \cdot \dot{m} = C_{cond} \cdot \dot{m}_{cond} \tag{12}$$

where:

$\dot{m}$ = mass flow rate of water in wet steam flow

$\dot{m}_{cond}$ = mass flow rate of condensate from wet steam

With the tracer concentrations known from test measurements, before and after condensation, steam wetness fraction (moisture) is represented by the ratio:

$$1 - x = \frac{\dot{m}}{\dot{m}_{cond}} = \frac{C_{cond}}{C_{wat}} \tag{13}$$

and the steam quality is determined by:

$$x = 1 - \frac{C_{cond}}{C_{wat}} \tag{14}$$

Throttle quality can be calculated from the quality and pressure of steam leaving the steam generator, and throttle pressure.

The moisture in the steam leaving the steam generator is the result of water carry-over. Thus, a tracer present in the steam generator water will also be found in the steam.

In non-reheat cycles, the tracer will finally be diluted in the total flow going back to the steam generator. By the application of the condensing method, steam generator exit moisture can be evaluated using equation (14) above.

In the case of reheat cycles, the error in the throttle moisture determination caused by plating out of the tracer in the reheaters is negligible, provided that the external moisture separator effectiveness is approximately 100%. However, there is the possibility of measuring the moisture carry-over of the steam generator during special tests with the reheaters out of service if agreed between the parties concerned.

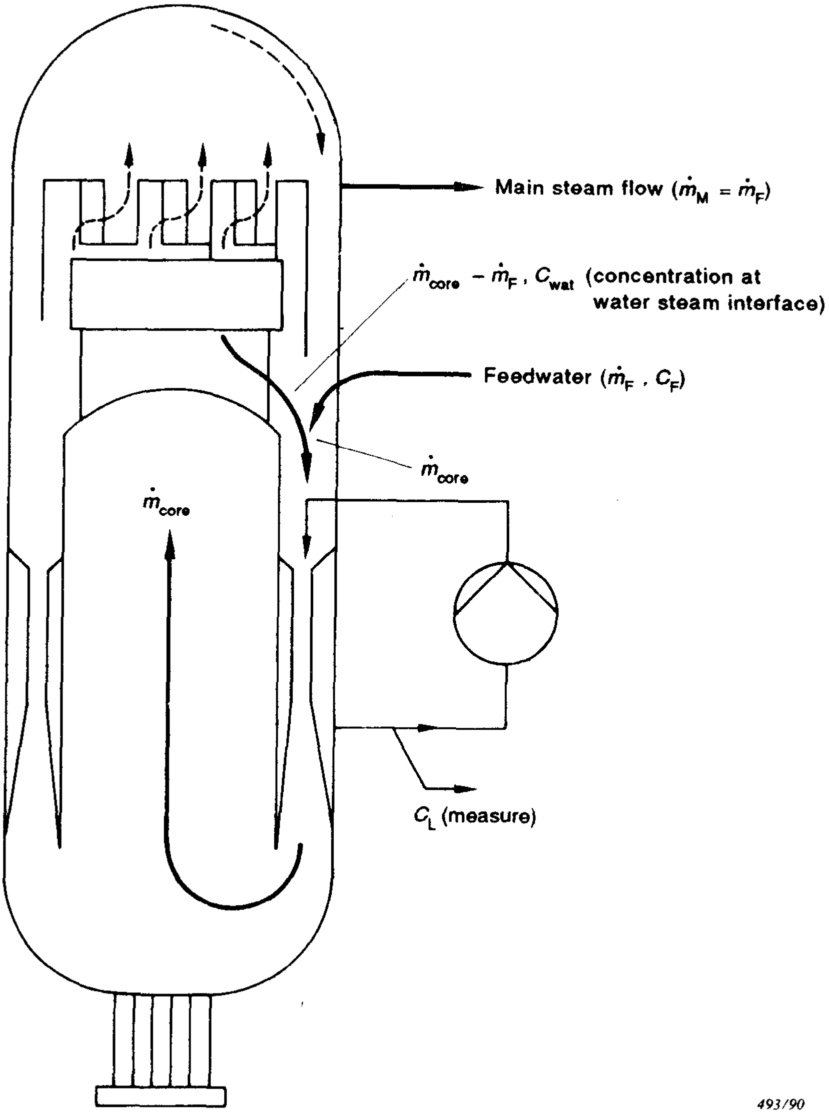
When the tracer concentration is measured at the bottom of the steam generator, the factor R (see Figures 4 and 5) has to be taken into account in the determination of the concentration C_{wat} which serves for the determination of the steam quality. Consequently, this procedure is less precise than the direct measurement of C_{wat}.

Determination of the concentration in the total flow, C_{cond} depends on the arrangement of the feedwater heaters. On cycles with cascading heaters, total flow usually exists at the discharge of the condensate pumps. On other cycles, if the demineralizers are bypassed during the test, C_{cond} will be the tracer concentration in the final feedwater. Another possibility is to calculate C_{cond} from a tracer flow balance. However, this calculation requires several flow and concentration measurements. In all cases, effects such as external tracer sources feeding into the cycle or losses of tracer (demineralizers) shall be taken into account.

The condensation method may also be used to determine wet extraction steam enthalpy. This method is particularly attractive if a suitable tracer is already present in the steam path. However, error analysis shows that accurate results can only be obtained on heaters without cascading drains.

With this method, extraction enthalpy is evaluated from an energy balance and a tracer balance around the heaters.

For the tracer balance, the concentration of the tracer in all flows to and from each heater shell side are needed. Sampling the heater drains for concentration measurement of the tracer is fairly easy, as this is only single phase flow. Sampling water out of the extraction line requires the same precautions as in the case of the injection method.


Main steam flow ($\dot{m}_M = \dot{m}_F$)
$\dot{m}_{core} - \dot{m}_F$, C_{wat} (concentration at water steam interface)
Feedwater ($\dot{m}_F$, C_F)
$\dot{m}_{core}$
$\dot{m}_{core}$
C_L (measure)
493/90

$$\dot{m}_M = \dot{m}_F$$

$$C_{wat}\,(\dot{m}_{core} - \dot{m}_F) + \dot{m}_F\,C_F = \dot{m}_{core}\,C_L \tag{15}$$

$$C_{wat} = \frac{\dot{m}_{core}\,C_L - \dot{m}_F\,C_F}{\dot{m}_{core} - \dot{m}_F} \tag{16}$$

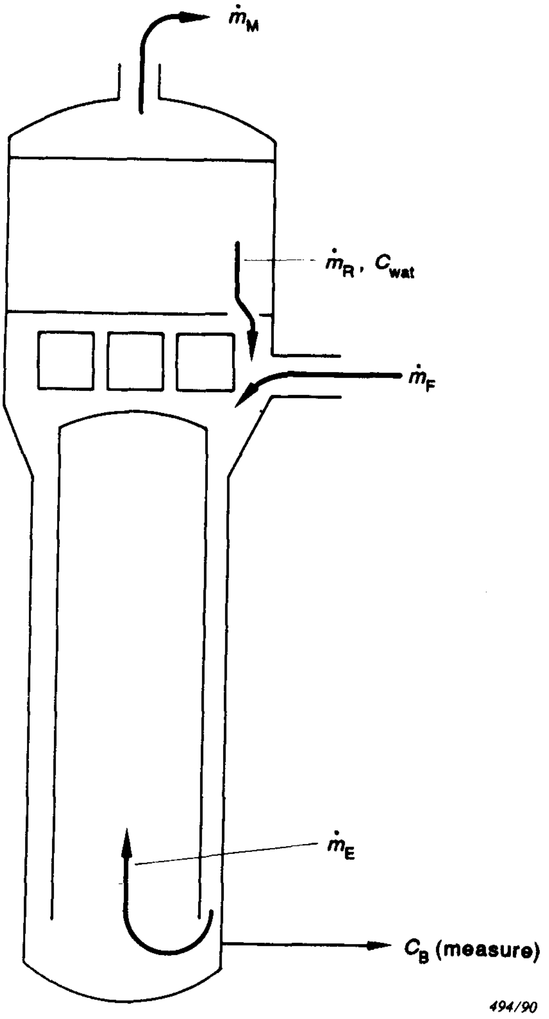
$$C_F \ll C_L \text{ and } \dot{m}_F < \dot{m}_{core}$$

$$C_{wat} = C_L \frac{\dot{m}_{core}}{\dot{m}_{core} - \dot{m}_F} = \frac{C_L}{1 - R} \tag{17}$$

where:

$$R = \frac{\dot{m}_F}{\dot{m}_{core}} = \frac{\dot{m}_M}{\dot{m}_{core}}$$

FIG. 4. — Throttle steam quality calculations for boiling water reactor.



C_{wat} = tracer concentration at water steam interface

C_B = blowdown tracer concentration

$\dot{m}_E = \dot{m}_R + \dot{m}_M$

$R = \frac{\dot{m}_M}{\dot{m}_E}$ or $\dot{m}_E = \frac{\dot{m}_M}{R}$

$$\dot{m}_R = \dot{m}_E - \dot{m}_M = \frac{\dot{m}_M}{R} - \dot{m}_M = (1 - R)\frac{\dot{m}_M}{R} \qquad (18)$$

$$C_{wat} \approx \frac{C_B\,\dot{m}_E}{\dot{m}_R} = \frac{C_B\left(\frac{\dot{m}_M}{R}\right)}{(1 - R)\frac{\dot{m}_M}{R}} = \frac{C_B}{1 - R} \qquad (19)$$

$$C_{wat} = \frac{C_B}{1 - R} \qquad (20)$$

FIG. 5. — Throttle steam quality calculations for pressurized water reactor.

4.7.4 *Constant rate injection method*

A water-soluble tracer of concentration C_{inj} is injected at a constant rate $\dot{m}_{inj}$ into the vapour-water flow where moisture is to be measured. The concentration C_{wat} is measured in the water phase downstream of the injection point after adequate mixing has taken place. For this condition the following material balance can be written:

$$C_o \cdot \dot{m} + \dot{m}_{inj} \cdot C_{inj} = (\dot{m} + \dot{m}_{inj} + \Delta\dot{m}) \cdot C_{wat} \tag{21}$$

or

$$\dot{m} = \frac{\dot{m}_{inj} \cdot (C_{inj} - C_{wat}) - \Delta\dot{m} \cdot C_{wat}}{C_{wat} - C_o} \tag{22}$$

where:

$\dot{m}$ = mass flow rate of water in vapour-water mixture at the sampling point

C_o = initial concentration in the water phase at the sampling point, before injection starts, due to natural amounts of tracer (background concentration)

$\Delta\dot{m}$ = change in water flow, (condensation of vapour due to injection of the cold tracer solution)

Normally $C_{wat} \ll C_{inj}$, $C_o \ll C_{wat}$ and $\Delta\dot{m} \ll \dot{m}$, so that the above equation is reduced to:

$$\dot{m} = \dot{m}_{inj} \cdot \frac{C_{inj}}{C_{wat}} \tag{23}$$

4.7.5 *Extraction enthalpy determined by constant rate injection method*

If the flow rate of the water phase in the extraction line to a feedwater heater is known, wet steam enthalpy can be calculated by energy balance around the heater.

The water flow rate can be measured with the constant rate injection method. Measuring flow rate and concentration of the tracer solution and maintaining a constant injection rate is comparatively simple. However, the tracer concentration in the phase downstream of the point of injection can be accurately determined only if the tracer is well mixed and a sample of the liquid phase can be obtained.

4.7.5.1 *Injection points*

For the sample to be truly representative, the tracer shall be uniformly distributed in the water phase. The injection point should, therefore, be located immediately downstream of the extraction flange of the turbine, and the sampling point should be close to the heater. A long run of pipe with several elbows will promote mixing. Use of a spray for injecting would be beneficial, but it may not be necessary.

4.7.5.2 *Sampling points*

Since any condensed vapour in the sample will falsify the result, care shall be taken in selecting the location of the sampling tap. At the conditions and velocities normally found in extraction lines, the water is not uniformly distributed over the cross-section, but agglomerated towards the pipe wall. This is favourable for water sampling, so that a simple wall tap should prove satisfactory. However, advantage should be taken of gravitational or centrifugal forces by locating the tap on the bottom of the pipe or on the outside at the exit of an elbow. Figure 6 shows a typical installation of injection and sampling points. In cases where the extraction line is very short, such as heaters installed in condenser necks, problems may be encountered in obtaining water samples. In such cases, these rules recommend that the water sample be taken from the heater drain.

In cycles with pumped ahead drains, the above method yields accuracy equal to the injection method without any additional instrumentation. In cycles with drains cascading to the condenser, the upstream drains shall be diverted directly to the condenser so as to yield comparable results. Such diversion necessitates designing such drain piping to the condenser to accommodate total cascade flow at full load. The diversion method should be employed prior to the heat rate test.

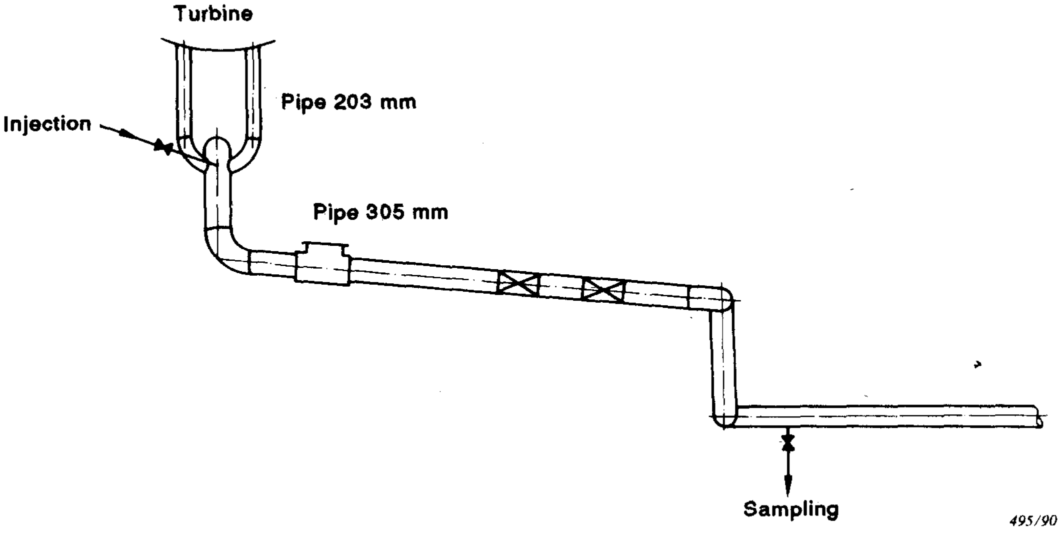


FIG. 6. — Typical installation of injection and sampling points.

4.7.5.3 *Sampling flow rates*

The sampling flow rate shall be adjusted so that entrainment and subsequent condensation of vapour is prevented. The maximum allowable sampling rate can be determined, for example, by analyzing the sampling stream for dissolved oxygen. Oxygen (20 to 30ppm) is naturally present in the steam from boiling water reactors as a result of radiolysis. The sampling flow rate shall be determined prior to the heat rate test. The oxygen content shall be measured for various sample flow rates and plotted as shown in Figure 7. The flow rate at which steam starts to entrain in the sampling is evidenced by a sharp rise in oxygen content. The validity of using oxygen or other suitable tracer as a means of tracing the vapour fraction is based on the distribution of oxygen between the liquid and vapour phases. At pressures less than 35bar, oxygen is almost entirely in the vapour phase.

In other types of plants, a suitable tracer, such as xenon-133, could be added.

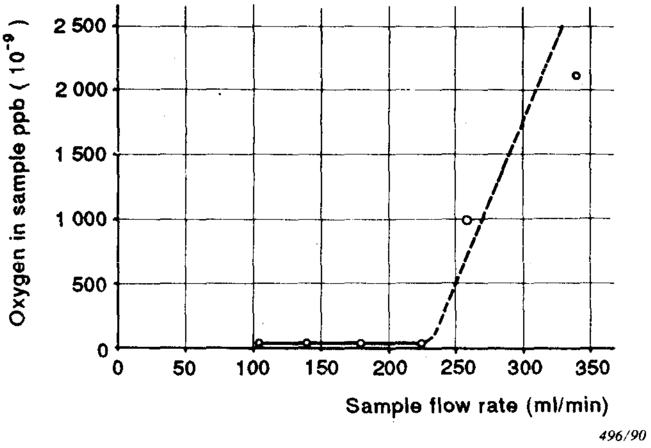


FIG. 7. — Oxygen content of sample stream.

4.7.6 *Tracers and their use*

For condensing and constant-rate injection methods to give accurate results, tracers shall meet the following criteria:

a) No hazard to operating personnel.

b) No harmful effects on materials employed.

c) Soluble in water, but essentially insoluble in steam (concentration in the steam phase of 10^{-6} can be attained with suitable tracer compounds).

d) Non-volatile.

e) Stable at the conditions existing in the turbine cycle, and not absorbed on internal surfaces (provided that the water is not evaporated completely).

f) Mixed completely and uniformly with all the water available at any instant.

Light water reactors of the present generation are designed for pressure and temperature levels which permit a number of tracers to meet these criteria. At higher pressure levels this may be more difficult. Furthermore, the problem of chemical contamination, which is tied together with the sensitivity and accuracy of the analytical methods available, has to be considered. Methods such as measurement of conductivity or the gravimetric methods may be too insensitive to produce the required accuracy at the concentration levels which can be tolerated in some nuclear system. Care shall be taken to prevent environmental contamination when tracers are used.

4.7.6.1 *Radioactive tracers*

Radioactive tracers are particularly attractive for application in nuclear power plants, where licensing required for possession and handling of radioactive materials presents no particular problem. Tracer concentrations of less than 1 in 10^9 can be accurately measured using gamma counting techniques. For steam cycles with very low radioactive background, the tracer activity concentration required for accurate testing is very small; however, the tracer should be a short-lived isotope to eliminate long-term contamination problems. Since it is not practicable to measure the concentration of recovered samples simultaneously it will be necessary to apply a correction to the measured concentration of each sample to account for isotope decay. One of the tracers which meets these criteria is 15-h sodium-24. If the tracer technique is used to determine throttle and extraction steam quality and heater leakage, there is some advantage in using three different radioactive tracers.

4.7.6.2 *Non-radioactive tracer*

A number of tracers are available. The sodium tracer technique is based on accurate and direct determination of the mass concentration of sodium in water samples. Sodium may be present in PWR cycles due to the use of sodium phosphates for secondary loop water chemistry control. The sodium tracer technique also preserves the non-radioactive status of the secondary loop in PWR cycles.

a) Flame photometry method

Sodium analysis may be performed with a flame spectrophotometer. Care shall be taken to prevent environmental sodium from contaminating a sample. Multiple determinations of sodium concentrations shall be made for each sampling point. Sodium concentration may also be determined from a continuously flowing sample.

b) Sodium ion electrode method

The sodium analyzer may be used for continuous monitoring of a flowing sample, as well as for analyzing separate samples.

The application of non-radioactive tracer techniques shall be consistent with regulations concerning the metallurgical safety of the reactor system and all other related equipment. Lithium salts may also be used in preference to sodium.

4.8 *Time measurement*

The time of test periods and observations may be determined by:

a) signals from a master clock or timekeeper;

b) observations of watches, which are to be synchronized before the test, by the individual observers.

A time measurement of high accuracy is necessary for each measurement employing an integrating meter. Individual stop watches or electric time measuring devices shall be employed. Special care shall be given to the accurate synchronization of the readings of the integrating meter and the time measuring devices.

4.9 *Speed measurement*

Speed can be measured by means of a counting mechanism and clock, a stroboscope, a frequency measuring instrument (mechanical or electrical) or a speed measuring instrument. Signal transducers and signal transformers are also accepted.

5. Evaluation of tests

5.1 *Preparation of evaluation*

From the instrument readings taken during the test, the test results according to 2.4 are calculated.

Prior to evaluating the measurements, a time interval which shall constitute the official test period shall be selected from the overall period during which readings for a test have been taken. This interval shall be equal to at least the period laid down in 3.8.3. During this period the provisions of 3.8.1 and 3.8.2 relating to the deviations of the operating conditions from the guarantee conditions and relating to the fluctuations in the operating conditions shall be satisfied.

Readings of all instruments, including integrating instruments, and corresponding time measurements shall be available for the start and the end of the selected test period (see 5.2.1).

In the event of disturbances encountered during a test, all instrument readings made during the disturbed period may by mutual agreement be eliminated according to 4.4.7. Evidence shall be furnished that the same steady-state conditions had existed before and after the disturbance. The total length of the remaining test periods shall comply with 3.8.3.

In the event of a temporary failure of a measuring instrument during the test, the missing readings may by agreement and where appropriate be deduced from other suitable instrument readings.

5.2 *Computation of results*

5.2.1 *Calculation of average values of instrument readings*

For further evaluation, the average value of the readings taken during the test period defined according to 5.1 is calculated for each measuring instrument.

For variables which influence the evaluation in a linear manner, the average value is the arithmetic mean value. The theoretically correct procedure for averaging the readings of a pressure difference across a flow-measuring device is to calculate the arithmetic mean value of the square roots of the readings. However, if the amplitude of readings is less than 10%, the maximum possible error introduced by not calculating the square root of the arithmetic mean of the readings is less than 0.1%.

If data from integrating instruments are compared with averaged readings of indicating instruments, it is important to ensure that they have been determined for the same time period. The first and last readings of the indicating instruments should then carry only half the weight in the averaging procedure.

5.2.2 *Correction and conversion of averaged readings*

The averaged readings are corrected to take care of all influences resulting from the instrumentation and converted to the necessary units considering:

a) instrument constants and zero corrections;

b) calibration corrections;

c) reference values of instrument readings (e.g., barometric pressure, ambient temperature);

d) any additional influences (e.g. water legs).

5.2.3 *Checking of measured data*

5.2.3.1 *Compatibility*

After computation of the measured data, such as pressures, temperatures, flow rates, etc., a thorough examination shall be made for serious errors, inconsistencies with the laws of physics, and general compatibility. If major discrepancies are detected, the cause and extent of which are unknown, the test run shall be repeated completely or to the necessary extent. Appropriate additional measurements shall be made for clarification. Instrument readings which are obviously incorrect shall be eliminated. Subject to mutual agreement of the parties interested in the test, such readings may be replaced by other instrument readings or by appropriate calculated or estimated values.

5.2.3.2 *Evaluation of multiple measurements*

Where results from several independent instruments are available for the same variable, an average value of these shall be calculated in a suitable manner, which considers the relative reliability of the individual measured values x_j by applying weighting factors γ_j.

The most probable figure for the average value of several results x_j is the weighted mean value $\tilde{x}$

$$\tilde{x} = \frac{\Sigma (x_j \gamma_j)}{\Sigma \gamma_j} \qquad (24)$$

Each weighting factor is calculated from the confidence limit V_{x_j} of the result x_j (see clause 7)

$$\gamma_j = \frac{1}{V_{x^2j}} \qquad (25)$$

In order to check the mutual compatibility of several measured values for the same variable with consideration of their calculated or estimated measuring uncertainty and for the justification of the averaging as indicated above, the application of a statistical method can be helpful; this is described in detail in Appendix C.

5.2.3.3 *Mass flow balances*

When several mass flow measurements are made in the main steam-water circuit during a test, several values can be calculated for a certain mass flow in the main circuit (e.g., the initial steam flow) by setting up mass flow balances.

For the establishment of these balance equations, it is further necessary to measure certain secondary mass flows as well as level changes in locations where water storage can occur.

The setting up of mass flow balances and the computation of the results is treated in detail in Appendix D.

5.2.3.4 *Leakages*

Leakages shall, as far as is practicable, be identified and eliminated before the test. If any identified leakages cannot be eliminated, their flow rates shall be measured or estimated. These estimates shall be included in the main or secondary flow rate calculations. Unidentified leakages, which result in a loss of working fluid, shall also have their flow rate(s) and positions(s) estimated and included, if considered necessary, in the main or secondary flow rate calculations.

The total unaccounted leakage during the test expressed in percent of full load initial steam flow shall not exceed 0.4 times the relative measuring uncertainty of the test result expressed as a percentage, otherwise the test can be accepted only by mutual agreement of the parties to the test.

5.2.4 *Thermodynamic properties of steam and water*

The steam and water tables and charts upon which the guarantees are based shall be used for the calculation of the test results. The name and edition of the tables and charts employed shall be stated in the report of the tests. The tables used shall be consistent with the International Skeleton Tables established at the Sixth International Conference on the Properties of Steam (ICPS) in 1963, and preferably be based on the 1967 IFC Formulation for Industrial Use that was approved at the Seventh ICPS in 1968, or the latest edition.

5.2.5 *Calculation of test results*

The thermal and/or thermodynamic efficiency, the initial steam flow capacity and the output have to be calculated in accordance with the definition of the guarantee values (see 2.4).

In accordance with 5.1 and 5.2, a definite value shall be established for each variable which shall be used exclusively in further evaluation.

For the determination of measuring uncertainty, see clause 7.

6. Correction of test results and comparison with guarantee

6.1 *Guarantee values and guarantee conditions*

The guarantee values with which the thermal acceptance test results shall be compared indicate the level of the turbine performance. Their definitions are given in 2.4. These values are dependent upon steam cycle terminal conditions, and upon the parameters of the feed-water heating system, so each of these shall be defined completely and clearly in the contract document to constitute the specified guarantee conditions. In many cases it is recommended that these data be given in the form of a heat-balance diagram containing all necessary information.

The test results are equally dependent on these same conditions and parameters, so any observed departure from the value upon which the guarantee is given becomes subject to a correction after the test (see 6.4).

If extraction or addition of steam and/or water is already included in the guarantee cycle, it is an additional specified guarantee condition.

6.2 *Correction of initial steam flow capacity*

For correction of the initial steam flow capacity to guarantee conditions, the following equation is applied:

$$\dot{m}_{1,\max,c} = \dot{m}_{1,\max,m} \cdot \frac{p_{1,g}}{p_{1,m}} \sqrt{\frac{p_{1,m} \cdot v_{1,m}}{p_{1,g} \cdot v_{1,g}}} \tag{26}$$

where:

$\dot{m}_{1,\max,m}$ = initial steam flow capacity measured during test with valves wide open

If the ratio of exhaust pressure to inlet pressure of the turbine is not sufficiently small, the use of the complete equation may be necessary:

$$\dot{m}_{1,\max,c} = \dot{m}_{1,\max,m} \cdot \frac{p_{1,g}}{p_{1,m}} \sqrt{\frac{p_{1,m} \cdot v_{1,m}}{p_{1,g} \cdot v_{1,g}}} \sqrt{\frac{1 - \left(\frac{p_{2,g}}{p_{1,g}}\right)^2}{1 - \left(\frac{p_{2,m}}{p_{1,m}}\right)^2}} \tag{27}$$

The exponents 2 of the pressure ratios are approximations of the exact value $\frac{1+k}{k}$.

6.3 *Correction of maximum output*

In the correction of the maximum output P_{max} to guarantee condition all the operating conditions influencing the initial steam flow, the enthalpy drop and its distribution in the different parts of the turbine and the efficiency values in the different parts of the turbine should be taken into account.

Corrections for all these parameters (particularly live steam condition, reheat temperature and condenser pressure) can be made with appropriate correction curves.

A complete correction can be made by appropriate recalculation of the thermal cycle.

An approximate correction formula assuming similar distribution of enthalpy drop and efficiency is:

$$P_{max,c} = P_{max,m} \cdot \frac{\dot{m}_{1,max,c}}{\dot{m}_{1,max,m}} \cdot \frac{\eta_{max,c}}{\eta_{max,m}} \tag{28}$$

For the determination of $\dot{m}_{1,max,c}$ see 6.2. The efficiency to be used shall be according to the formulation of the guarantees, the thermal efficiency η_t or the thermodynamic efficiency η_{td}.

6.4 *Correction of thermal and thermodynamic efficiency*

If any operating conditions according to 6.1 differ during the test from the guarantee conditions, the thermal or thermodynamic efficiency, or heat rate or steam rate have to be corrected before they can be compared with the guarantee values. In order to keep the correction small, the operating conditions during the test shall be as close as practicable to the specified guarantee conditions. The maximum permissible deviations for the most important operating parameters are stated in 3.8.2.

The correction for deviations from specified guarantee conditions can be divided into three categories.

Category 1 comprises corrections for deviations of the terminal operating conditions of the turbine itself from specified guarantee conditions, such as (see also 6.7):

a) initial steam pressure,

b) initial steam temperature,

c) reheat steam temperature,

d) intitial steam quality,

e) reheater pressure drop,

f) turbine-exhaust pressure or condenser cooling water temperature and flow,

g) moisture-separator effectiveness, if not supplied with the turbine,

h) speed.

Category 2 comprises variables which primarily affect the feed-heating system. This category includes corrections, such as the following (see also 6.7):

a) feed-heater terminal differences,

b) extraction-line pressure drops,

c) system water-storage changes and make-up flow,

d) enthalpy rise through condensate and feed-pumps,

e) condensate undercooling in condenser,

f) spray-attemperator flows for boiler, *g)* different configuration of feed-heating system (for example, heater out of service).

Category 3 corrections concern the generator operating conditions. They are independent of the remainder of the plant and simple to determine. If, however, the heat equivalent of the generator losses is transferred into the feed-water, they are of greater consequence and most conveniently treated in Category 2:

a) generator power factor,

b) voltage,

c) hydrogen pressure.

6.5 *Definition and application of correction values*

For correction purposes, it is normally assumed that the influences of the various operating parameters on the test result are mutually independent of each other. Therefore the correction values are determined individually according to the deviations of the individual operating parameters and are then combined to form a total correction.

Correction values are factors defined as follows:

$$F = \frac{\eta_c}{\eta_m} \tag{29}$$

where:

η_c = corrected efficiency based on specified guaranteed conditions

η_m = measured efficiency

The individual correction factors F_1, F_2, F_3... are then combined to give a total correction factor F_{tot}.

$$F_{tot} = F_1 \cdot F_2 \cdot F_3 \ldots$$

The corrected efficiency value is then:

$$\eta_c = \eta_m \cdot F_{tot} \tag{30}$$

For all parameters, where the correction is load dependent, correction factors have to be determined for each guarantee load point.

The correction of the test result to specified guarantee conditions can also be effected by a complete recalculation of the turbine and its thermal cycle usually by means of a computer program with due consideration of the characteristics of the turbine and plant components and the operating conditions during the test. This method yields a complete correction directly, without the use of individual correction factors and also takes into account to a large extent the interdependence of the different variables.

It may also be convenient to make only the corrections of category 2 by recalculation, since correction curves for these variables may not be readily available. The category 1 corrections are then made by means of correction curves.

6.6 *Correction methods*

The determination of the correction method and of the necessary values and curves depends on the method to be applied for the guarantee comparison (see 6.8).

a) If the guarantee comparison is made for constant initial steam flow, for each variable its influence on the thermal or thermodynamic efficiency has to be considered for the correction.

b) If the guarantee comparison is made for constant valve position of the turbine, the influence on both efficiency and output of each variable has to be determined.

The correction and comparison procedure to be applied should be agreed upon by the parties interested in the test sufficiently early before the test to allow the necessary preparatory work to be completed. The manufacturer's correction curves should be supplied within an agreed period, usually three months before the test, unless agreement to the contrary has been made in the contract.

The correction method described in 6.6.1 is recommended, in particular, for turbines with complicated cycles (turbines with steam extraction or bleeding for other purposes, with secondary steam, etc., or with complex feed-water heating cycles) and for turbines with many cycle corrections. If suitable means of calculation are applied, this method gives corrections which are more exact, complete and useful than corrections according to the method described in 6.6.2. Other suitable correction methods may also be applied by agreement.

6.6.1 *Correction by heat balance calculation*

The correction may be made by calculating a heat balance to correct the test cycle and the test operating conditions to the guarantee cycle and guarantee operating conditions. That is, the turbine test efficiencies may be corrected according to the specified conditions (for the influence of operating conditions, if necessary). These corrected test efficiencies may be used to calculate a "corrected test" cycle which can be compared with the guarantee cycle.

Alternatively, the correction may be made by calculating a heat balance to correct the guarantee cycle and the guarantee operating conditions to the test cycle and test operating conditions. The turbine guarantee efficiency may be corrected according to the test conditions (for the influence of operating conditions, if necessary). This corrected test efficiency may be used to calculate a "corrected guarantee" cycle which can be compared with the test cycle.

For turbines using superheated steam, the first method is usually applied, whereas for turbines using wet steam the second method may be easier to perform correctly.

For the correction by recalculation of a complete heat balance under test conditions, a calculating method (program) is necessary. This correctly takes into account the influence of the test conditions on the performance of the turbine and the plant components.

If the program which had been used for the calculation of the contract guarantee cycle cannot be used, the following method can be applied:

First a thermal efficiency value at guarantee conditions η'_g is determined with the available program, which may deviate slightly from the guaranteed efficiency η_g because of minor differences between the programs and data used for the determination of the guarantee values and for the recalculation. Then a thermal efficiency η'_m under test conditions is calculated with the same program and the same performance characteristics of turbine and plant components.

The measured thermal efficiency corrected for test conditions η_c is then:

$$\eta_c = \eta_m \frac{\eta'_g}{\eta'_m} \qquad (31)$$

The correction by calculation may be limited to the variables of Category 2. The corrections for variables of Categories 1 and 3 are made by use of correction curves. If the recalculation of the cycle for Category 2 corrections results in changes in the values of any parameters in categories 1 or 3, then these changed values shall be used when applying corrections for them from correction curves.

Both principles described in 6.6 may be applied for correction by heat balance calculation. The method of guarantee comparison is to be chosen accordingly (see 6.8).

All necessary data and calculation methods shall be made available to the parties to the test sufficiently early before the acceptance test.

6.6.2 *Correction by use of correction curves prepared by the manufacturer*

For case 6.6.*a)* the correction curves give values of the correction factor F_η for the efficiency as a function of the different variables. For case 6.6*b)* curves for the correction factors F^*_η for the efficiency - different from those for case 6.6*a)* - and for the correction factors F^*_p for the output as a function of the different variables are necessary.

If a correction factor depends also on initial steam flow (case 6.6*a)*), or output (case 6.6*b)*), several correction curves for different loads, preferably for the guarantee loads, are necessary.

It shall be clearly indicated on the correction curves the method described in 6.6 for which they have been prepared and have to be applied.

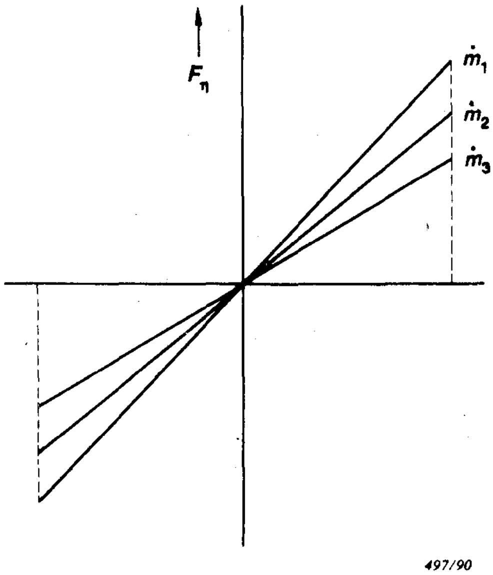


FIG. 8a. — Correction according to 6.6*a)*.

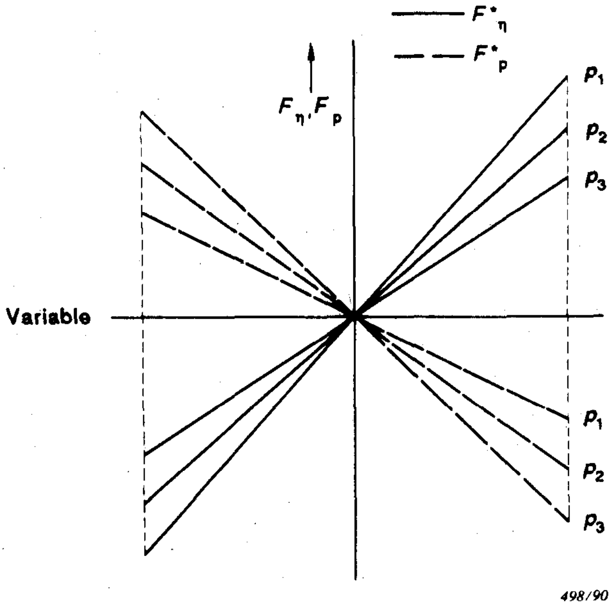


FIG. 8b. — Correction according to 6.6*b)*.

FIG. 8. — Typical correction curves.

6.6.3 *Tests to determine correction values*

It is possible that the calculation of the correction factors referred to in 6.6.2 may not achieve a satisfactory standard of accuracy. In this event, and especially in relation to the exhaust pressure, it is recommended that special tests be carried out to determine the correction factors.

All tests of the same series in which one quantity is varied in a controlled way (e.g., the exhaust pressure by admitting air to the condenser) shall be run on the same day, by the same personnel and with the same apparatus. A sufficient interval of time shall be allowed between tests of the same series for stable conditions to be established.

6.6.4 *Correction by use of generalized correction curves*

The typical generalized correction curves in Appendix E may be used by agreement for the correction of thermal efficiency test results of a wide range of condensing turbines for the more important variables with sufficient accuracy, if no other means are available. They can be used for the procedure according to 6.6*a)* only.

6.7 *Variables to be considered in the correction*

Corrections of the test results may be made only for deviations from the guarantee conditions at the inlets to and/or outlets from the system, used as a basis for the definition of the guarantee, or for deviations inside this system, which are beyond the responsibility of the turbine supplier. In an individual case "system" can mean:

a) turbine only;

b) turbine and feed-water cycle with or without pumps;

c) turbine, feed-water cycle, condenser;

d) turbine, feed-water cycle, condenser and/or driven machine.

6.7.1 *Turbines with regenerative feed-water heating*

When applying corrections to the results of thermal efficiency or heat rate tests on turbines with regenerative feed-water heating, deviations of the following from the guarantee conditions have to be considered:

Category 1:

a) initial steam pressure,[1]

b) initial steam temperature,

c) reheat steam temperature,

d) pressure drop in the reheater,

e) exhaust pressure or condenser cooling water inlet temperature and flow rate,

f) final feed-water temperature, if required,[2]

g) speed.

Additional variables for turbines using wet steam:

h) initial steam quality,

---

[1] Deviations of initial steam pressure from the specified value, resulting from a deviation of the steam flow capacity of variable pressure turbines from the specified value within the limits stipulated in the contract, shall be considered in the correction.

[2] Deviations of final feed-water temperature from the specified value, resulting from a deviation of the top heater terminal difference or steam extraction pipe pressure drop from the specified value within the limits stipulated in the contract, shall be considered in the correction.

i) steam quality at the outlet of the moisture separator,[1]

j) terminal difference of reheater,[1]

k) pressure drop in moisture separator and reheater.[1]

Category 2:

a) inlet temperature of condensate into feed-heating system (undercooling of condensate in condenser, influence of hydrogen cooler, oil cooler and ejector-steam condenser),

b) water flow for spray attemperation in the superheater and the reheater unless this water is collected at the outlet of feed-heating system,

c) terminal difference of feed-heaters, condensate coolers and desuperheaters,[2]

d) pressure drop in extraction lines,[2]

e) enthalpy rise of feed-water in the feed-pump,

f) modifications of the configuration of the feed-heating system (e.g. total or partial by-pass of heaters),

g) make-up water,

h) level in feed-water reservoirs,

i) steam-ejector steam flow,

j) steam supply and steam inlet conditions of boiler feed-pump turbine,

k) any other heat additions or extractions (see also 6.4),

l) initial steam flow,[3]

m) pressure drop in regulating valve(s) in throttling position[2],

n) any further deviations from guarantee conditions.

Category 3:

a) generator power factor,

b) voltage,

c) generator cooling gas pressure and purity.

6.7.2 *Turbines which have no provision for the addition or extraction of steam after partial expansion*

When applying corrections to the results of thermal efficiency or steam rate tests on turbines which have no provision for the addition or extraction of steam after partial expansion, deviations of the following variables from the guarantee conditions have to be considered:

Category 1: *a)* initial steam pressure,
b) initial steam temperature,

---

[1] Only if this equipment does not form part of the scope of the contract.

[2] Deviations of final feed-water temperature from the specified value, resulting from a deviation of the top heater terminal difference or steam extraction pipe pressure drop from the specified value within the limits stipulated in the contract, shall be considered in the correction.

[3] Deviations of initial steam pressure from the specified value, resulting from a deviation of the steam flow capacity of variable pressure turbines from the specified value within the limits stipulated in the contract, shall be considered in the correction.

c) exhaust pressure or condenser cooling water inlet temperature and flow rate,

d) speed.

Category 2: *a)* initial steam flow,[1]
b) pressure drop in regulating valve(s) in throttling position.[2]

Category 3: *a)* generator power factor,
b) voltage,
c) generator cooling gas pressure and purity.

6.7.3 *Turbines with steam extraction for purposes other than feed-water heating (extraction turbines)*

When applying corrections to the results of thermal efficiency or steam rate tests, deviations of the following variables from the guarantee conditions have to be considered in addition to those indicated in 6.7.2.

- Extraction pressure.
- Extraction flow.[3]
- Pressure drop in the extraction regulating valve(s) in throttling position.[4]
- Any other deviations from guarantee conditions.

If a feed-heating system is connected to the turbine, relevant corrections in accordance with 6.8.1 shall also be made.

6.7.4 *Other types of turbine*

This category includes all types of single or multiple extraction turbines, and turbines with secondary initial steam, etc., not classified in the previous sub-clauses

Because of the great number of possible variables for correction, an exhaustive list cannot be given. Many of the important variables, however, are listed in 6.8.1 to 6.8.3. Further relevant variables may have to be determined for each case.

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[1] A correction for differing initial steam flow is necessary only if correction and guarantee comparison are made to a guarantee point, not to a locus curve (see 6.8.1).

[2] The guarantee values are specified normally for valve points (see 3.6.1) and tests are conducted at valve points also with due consideration for the limits of deviations from guarantee conditions specified in Table I. A correction is not then necessary. If a throttle point is guaranteed (e.g. full arc admission turbine with throttle regulation at part load) the test shall be made with the throttling pressure drop specified in the guarantee conditions, or a correction has to be made.

[3] Deviations of initial steam pressure from the specified value, resulting from a deviation of the steam flow capacity of variable pressure turbines from the specified value within the limits stipulated in the contract, shall be considered in the correction.

[4] A correction for differing extraction steam flow is necessary only if correction and guarantee comparison are made to a guarantee point, not to a locus curve (see 6.8).

6.8 *Guarantee comparison*

The comparison of corrected test results with the guarantee values specified in the contract shall be made in conformity with the correction method chosen (see 6.6).

Taking into account the measuring uncertainty of the test result, the guarantee is considered fulfilled if:

$$\eta_c + \tau_\eta \cdot \eta_m \geqslant \eta_g \tag{32}$$

unless otherwise stated in the contract (see also 7.5).

When *n* guarantee points are given, the guarantee is considered fulfilled if:

$$\Sigma_1^n \left(\eta_{ci} + \tau_{\eta i} \cdot \eta_{mi} - \eta_{gi}\right) \geqslant 0 \tag{33}$$

unless otherwise stated in the contract.

If the different guarantee points are weighted in the contract, the weighting factors shall be applied correspondingly in the guarantee comparison.

6.8.1 *Guarantee comparison with locus curve*

If several guarantee points with identical definition (normally “valve points“) are given, a locus curve of the guarantee values can be established.

If the test results have been corrected according to 6.6*a)*, the locus curve shall be established as a function of initial steam flow $\dot{m}_1$ and the corrected test result η_c compared with the value η_g of the guarantee curve at the measured initial steam flow $\dot{m}_{1,m}$ (see Figure 9a).

If the test results have been corrected according to 6.6*b)*, the locus curve shall be established as a function of output P and the corrected test result η_c compared with the value η_g of the guarantee curve at the guaranteed output P_g (see Figure 9b).

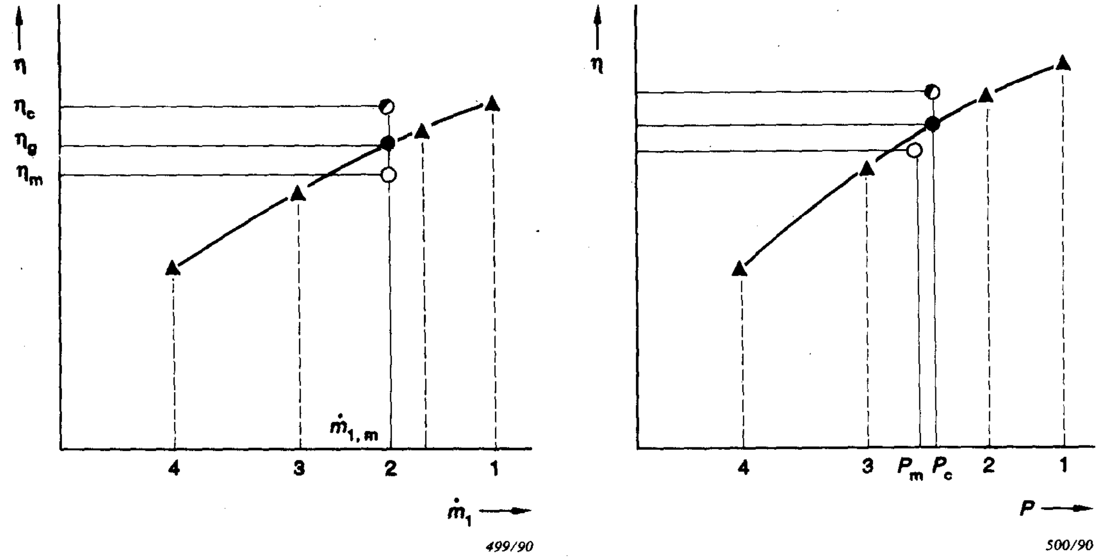


FIG. 9a. — Locus curve of initial steam flow.

FIG. 9b. — Locus curve as function of output.

▲ Guaranteed points
○ Measured points
◐ Corrected points
● Guaranteed point on locus curve

FIG. 9. — Guarantee comparison on locus curve.

6.8.2 *Guarantee comparison with guarantee point*

For comparison with a single guarantee value, a test result corrected according to 6.6*a)* has to be corrected further for the influence of the difference between measured initial steam flow and guaranteed steam flow.

If the test result has been corrected according to 6.6*b)*, only the difference between the corrected output and the guarantee output shall be considered for the further correction.

6.8.3 *Guarantee comparison for turbines with throttle governing*

If guarantees at partial loads are given for turbines with throttle governing, the amount of throttling in the control valves shall also be stated. The test result then has to be corrected also for the difference in throttling between guarantee condition and test conditions in a way which is in accordance with the method chosen in 6.6. Then the guarantee comparison can be made according to 6.8.1 or 6.8.2.

Alternatively, the corrected test value can be compared with the value of the guarantee locus curve at the same percentage value of initial steam flow capacity or of corrected maximum output with valves wide open. The necessary value of the initial steam flow capacity or maximum output of the tested turbine can most conveniently be determined by a test with valves wide open.

6.8.4 *Guarantee comparison for extraction turbines*

If an extraction diagram is guaranteed or can be established from the guarantee values for an extraction turbine, the method according to 6.6.*a)* can be applied. The extraction line for the measured extraction steam flow is determined by interpolation and used as the guarantee locus curve. Correction and guarantee comparison are then made with constant initial and extraction steam flow.

6.9 *Deterioration of turbine performance (ageing)*

The test shall be conducted as soon as practicable after initial start up (see 3.3.1) or after an inspection provided any deficiency in the unit affecting performance has been corrected. If these conditions cannot be met and comparison measurements for determination of the condition of the steam turbine according to 3.5 have not been made, the actual overall deterioration cannot be determined.

If there is reasonable assurance that the unit is free of partial damage and deposits, an average value of the deterioration by ageing (see 3.4.2) may be agreed by the parties to the test and taken into account in the comparison of the test results with guarantee. If no special agreement exists, the following rule, which represents average deterioration for fossil fuel fired units, is given for guidance on the action to be taken:

Allowance for average deterioration by ageing:

Turbine rating P	Time between first synchronization and test[2]		
	2 to 12 months	12 to 24 months	
≤ 150 MW	0.1	0.06	% per month
> 150 MW	$0.1\sqrt{\frac{150^{1)}}{P}}$	$0.06\sqrt{\frac{150^{1)}}{P}}$	% per month

[1] P in MW.

[2] Periods during which the turbine casings are open will not be considered.

7. Measuring uncertainty

7.1 *General*

The measurement of each quantity entering into the computation of the test result is liable to some degree of error which depends on the quality of the measuring instruments and the conditions of the measurement. The test result is subject to a degree of uncertainty depending on the combined effect of all the errors of measurement.

A short statistical definition of measuring uncertainty and a justification for the application of statistical methods under the special circumstances of thermal acceptance tests are given in Appendix F.

According to Appendix F, the measuring uncertainty of a variable is established as a confidence limit of the total measuring error for a statistical probability of $P = 95\%$. These confidence limits for individual measurements cannot be determined from the readings of an acceptance test.

Confidence limits can be determined as follows:

a) from the measuring recommendations and standards;

b) from the accuracy classes (error limit) indicated by the instrument manufacturer (e.g. Bourdon gauges, wattmeters);

c) from the calibration accuracy of a measuring instrument (e.g. signal transducer) as stated in the calibration certificate;

d) from the influence of an unavoidable installation error;

e) from general measuring experience (e.g. differential pressure measurement by U-tube manometer).

Measurements with calibrated instruments, where the average of the readings is corrected for instrument deviations as determined by the calibration, may still be influenced by errors due to different conditions during the measurement and calibration. These errors are also related to the accuracy class of each instrument. Allowance — if necessary — should be made for these errors by taking into account a part between 0 and 100% of the accuracy class for the determination of the total uncertainty according to the general error propagation laws.

In the case of multiple measurement (see 5.2.3.2), the measuring uncertainty $V_{\bar{x}}$ of the weighted average value $\bar{x}$ of several independent measurements x_j of the same variable x will be approximately:

$$V_{\bar{x}} = \pm \frac{1}{\sqrt{\Sigma\,(1/V_{xj})^2}} \qquad (34)$$

The methods for the determination of uncertainties for individual variables and results are given in the following sub-clauses.

Data on measuring uncertainties according to general experience are provided in clause 4, Table II.

7.2 *Determination of measuring uncertainty of steam and water properties*

7.2.1 *Pressure*

In pressure measurement with deadweight gauges, Bourdon tube type gauges and transducers, the measuring uncertainty is determined by the accuracy class of the instruments and the error limits of the calibration. In pressure measurements with liquid manometers the measuring uncertainty depends, inter alia, on the fluctuations of the column, the shape of the meniscus and the quality of the means for reading the liquid level.

Measuring accuracy is also influenced by the design and location of the pressure tapping, the ambient temperature and vibrations.

7.2.2 *Temperature*

In temperature measurements with liquid-in-glass thermometers, the error limit of calibration shall be deemed to be the measuring uncertainty of the sensing element. In temperature measurements with thermocouples or resistance thermometers, the measuring uncertainty is established from the error limit of the emf. curve or the error limits of the measuring resistance (both of which can be limited by calibration) and from the accuracy classes of the measuring instruments (potentiometer, digital voltmeter, etc.).

The measuring uncertainty of temperature measurements is liable to be considerably increased by inappropriate installation of sensing elements, uncertainty of the temperature correction of the column of a liquid-in-glass thermometer, inaccurate cold junction temperature, differing terminal temperature, errors due to non-uniform temperature distribution or to flow disturbances.

7.2.3 *Enthalpy and enthalpy difference*

The enthalpy is usually determined from the instrument readings for pressure and temperature. The uncertainties of these quantities are therefore included in the uncertainty of the enthalpy. Further allowance shall be made for the tolerance R_h of steam table values for the enthalpy. The uncertainty of an enthalpy value is:

a) Superheated steam

$$V_h = \pm \sqrt{\left(\frac{\delta h}{\delta T} V_T\right)^2 + \left(\frac{\delta h}{\delta p} V_p\right)^2 + R_h{}^2} \qquad (35)$$

b) Wet steam

$$V_h = \pm \sqrt{\left[(1-x)\frac{\delta h'}{\delta p_{sat}} + x\frac{\delta h''}{\delta p_{sat}}\right]^2 \cdot V^2_{p_{sat}} + (h'' - h') \cdot V^2_x + R^2_{h'} + R^2_{h''}} \qquad (36)$$

p_{sat} can be replaced by t_{sat}.

In establishing the uncertainty of enthalpy differences, the steam table tolerance cannot always be applied in full. Three cases shall be distinguished:

a) Isobaric heat addition with change of phase, for example in a steam generator between the enthalpy values h_i and h_j

$$V_{\Delta h} = \pm \sqrt{V^2_{h_i} + V^2_{h_j}} \qquad (37)$$

where V_{h_i} and V_{h_j} are calculated according to equation (35).

b) Isobaric heat addition without change of phase (e.g. in reheating) between the enthalpy values h_i and h_j

$$V_{\Delta h} = \pm \sqrt{V^2_{h_i} + V^2_{h_j} + \left[\left(\frac{R_{h_i}}{h_i}\right)^2 + \left(\frac{R_{h_j}}{h_j}\right)^2\right](h_i - h_j)^2} \qquad (38)$$

where R_{h_i} and R_{h_j} represent the steam table tolerance of h_i and h_j.

The uncertainty values V_{h_i} and V_{h_j} are established in this case without the steam table tolerance, i.e. using $R_h = 0$ in equation (35).

c) Isentropic expansion in a steam turbine (enthalpy drop)

$$V_{\Delta h} = \pm \sqrt{V^2_{h_i} + V^2_{h_j} + (h_i - h_j)^2 A^2(s)} \qquad (39)$$

The uncertainty values V_{h_i} and V_{h_j} shall be established as in case *b)* with $R_h = 0$ from equation (35). To determine V_{h_i} use $V_{t_i} = 0$. The factor $A_{(s)}$ is given as a function of the entropy s in Figure 10.

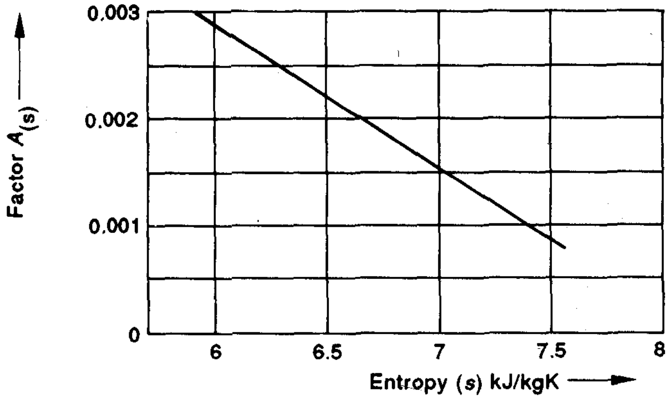


FIG. 10. – Correction factor for steam table tolerance.

7.3 *Calculation of measuring uncertainty of output*

7.3.1 *Electrical measurement*

According to the law of error propagation, the uncertainty of the measurement of the electrical power is in general calculated as the geometric summation of the individual measurement uncertainties of the transformers, the wattmeters, the series resistors, etc.

When curves or tables of the detectable systematic errors are available (from calibration of any part of the measuring equipment), these shall be used to correct the electrical power that has been measured (see 4.2.6, 4.2.7). The individual uncertainty limits as indicated in the certificates of the calibrations shall be decisive in establishing the measurement uncertainty of the electrical power (see also 7.1).

Where no details of detectable systematic errors are available, the calculation of the uncertainty of the total power shall be based on the accuracies class of the individual items of the measuring equipment.

In such cases, it is important to note the dependency of the accuracy class of the transformers on the values of current and voltage.

The accuracy class of a wattmeter shall be related to the actual scale value of the reading.

Practical experience has shown that the transforming ratio errors, as well as the phase-angle errors of equal transformers in parallel phases, are not fully independent. This also applies to the calibration of transformers. Therefore, in polyphase measurements (two-wattmeter or three-wattmeter method) the summation of errors is partially arithmetical.

In Appendix G, the separation of geometric and arithmetical errors and their summation is given in detail. It includes a number of formulae to calculate the measurement uncertainty of electrical power for two- and three-phase measurements, with wattmeters and watthour-meters, respectively.

7.3.2 *Mechanical measurement*

According to the law of error propagation, the measuring uncertainty is determined from the measuring uncertainties for speed and torque. The measuring uncertainty of the torque shall be established on the basis of the measuring procedure and the error limits of calibration.

7.3.3 *Additional uncertainty allowance because of unsteady load conditions*

If, during the test period, the output fluctuations above and below the mean value exceed ±5% (see 3.8), rejection of the test may be avoided by agreeing on an additional uncertainty allowance to compensate for the additional measurement errors. It depends on the magnitude of the fluctuations and can therefore be determined as follows:

the absolute values of the deviations of the individual output readings from the mean value are averaged.

$$\tau_{\Delta P} = \frac{2\,(\text{mean absolute deviation}) - 1}{6}\ (\%) \qquad (40)$$

$\tau_{\Delta P}$ is to be added arithmetically to the total uncertainty of the output measurement. The allowance $\tau_{\Delta P}$ is applicable only if the mean absolute deviation is greater than 0.5%.

7.4 *Determination of measuring uncertainty of mass flow*

7.4.1 *Measuring uncertainty of mass flow measurements*

The measuring uncertainty depends on the fluctuations of the reading, the shape of the meniscus and the quality of the cursor or other device which assists reading of the mercury level.

Non-detectable systematic errors can occur if the column cannot be observed directly (due to dirt in the sealing liquid, clinging of droplets or the influence of extraneous fields on a magnetic instrument).

Due to changes in liquid levels in storage volumes during tests, the reading of differential pressure measuring instruments may be subject to slow but considerable fluctuations. Such slow fluctuations shall not be taken into account in establishing the uncertainty of a differential pressure measurement.

The uncertainty of mass flow measurement is then determined for sharp-edged orifice plates or wall tap nozzles according to ISO 5167. If the differential pressure device has been calibrated, the uncertainty of the expansion coefficient may be estimated from the calibration conditions.

7.4.2 *Measuring uncertainty of multiple measurements of primary flow*

If primary flow has been determined by multiple measurements or by application of mass flow balances (see 5.2.3.3), their mutual compatibility has to be checked (see 5.2.3.1). In the case of incompatibility, all pressure difference devices which can be inspected (see 4.3.2.3) shall be checked for condition and size. A weighted average value is calculated from the measured values of all pressure difference devices which have been found in good order, and the measured values of all pressure difference devices without possibility of inspection are checked for compatibility with this average value (see 5.2.3.1).

An incompatible value should either be eliminated from further evaluation or its uncertainty value increased until compatibility is achieved. The decision has to be made with due consideration to the possibilities of imperfect cycle conditions and/or imperfections of the differential pressure device.

The possibility of errors at several uninspected differential pressure devices due to the same basic reason (unsharpness of edges or orifices due to wear, damage due to blowing out of pipeline, or chemical cleaning, etc.) should be duly considered in the evaluation.

If measured values of inspected differential pressure devices which have been found in good condition are not compatible and their uncertainty values have been determined correctly, an evaluation as multiple measurement according to 5.2.3.2 is not possible. The reasons for incompatibility are to be investigated further.

The value of the primary flow is calculated from the remaining compatible measured values according to 5.2.3.2. Its uncertainty value is determined according to 7.1.

7.4.3 *Uncertainty allowance for cycle imperfections*

To allow for imperfect conditions of the water-steam-cycle (unknown secondary flows, imperfect isolation, etc.), an additional uncertainty allowance equal to 50% of the part of the total unaccounted-for leakages, which enters into the further evaluation of the initial steam flow (see 5.2.3.4), is added arithmetically to the measuring uncertainty of the primary flow or the average uncertainty of the average primary flow (see 7.4.2).

7.5 *Calculation of measuring uncertainty of results*

7.5.1 *General*

The measuring uncertainty of the results is calculated by application of the law of error propagation from the uncertainties of the measured values according to Appendix F with due consideration of eventual interdependences of the variables and their uncertainties.

7.5.2 *Measuring uncertainty of thermal efficiency*

For a non-reheat condensing turbine plant, the thermal efficiency η_t is defined in 2.4.

Its relative measuring uncertainty is:

$$\tau_{\eta t} = \pm \sqrt{\tau^2_{\dot{m}_1} + \tau^2_{\Delta h_{11,1}} + \tau^2_P} \tag{41}$$

For a single-reheat condensing turbine plant, the thermal efficiency is defined in 2.4.

Normally, the reheater mass flow $\dot{m}_3$ is not measured independently, but calculated with the aid of secondary mass flows from the corrected initial steam mass flow. In the present case this relationship shall be introduced in the equation for the thermal efficiency before the measuring uncertainty is calculated.

In the example given in appendix D, the reheater mass flow is established from the equation:

$$\begin{aligned} \dot{m}_3 &= \tilde{\dot{m}}_1 - \dot{m}_{A5} - \dot{m}_{p1} - \dot{m}_{p2} - 0.5\,\dot{m}_{V3} \\ &= \tilde{\dot{m}}_1 - \Sigma\,\dot{m}_j \end{aligned} \tag{42}$$

which, inserted in equation (2), gives the thermal efficiency:

$$\eta_t = \frac{P}{\tilde{\dot{m}}_1 (\Delta h_{11,1} + \Delta h_{2,3}) - \Delta h_{2,3} \Sigma\,\dot{m}_j} \tag{43}$$

If, according to 5.2.3.3, the remaining dependences are ignored to simplify the evaluation, the relative measuring uncertainty of the thermal efficiency is:

$$\tau_{\eta t} = \pm \sqrt{A + B + C + \tau^2_{P}} \qquad (44)$$

with:

$$A = \left(\frac{\tilde{\dot{m}}_1 \cdot \Delta h_{11,1}}{Q_{tot.}}\right)^2 (\tau^2_{\tilde{\dot{m}}_1} + \tau^2_{\Delta h_{11,1}})$$

$$B = \left(\frac{\tilde{\dot{m}}_1 \cdot \Delta h_{2,3}}{Q_{tot.}}\right)^2 (\tau^2_{\tilde{\dot{m}}_1} + \tau^2_{\Delta h_{2,3}})$$

$$C = \Sigma_j \left[\left(\frac{\dot{m}_j \cdot \Delta h_{2,3}}{Q_{tot.}}\right)^2 (\tau^2_{\dot{m}_j} + \tau^2_{\Delta h_{2,3}})\right]$$

$Q_{tot.}$ being the total heat flow and $\dot{m}_j$ being the secondary mass flow according to equation (42).

7.5.3 *Measuring uncertainty of thermodynamic efficiency*

The thermodynamic efficiency η_{td} of a turbine is defined in 2.4. For the most simple case, the relative measuring uncertainty is:

$$\tau_{\eta td} = \pm \sqrt{\tau^2_{\dot{m}_1} + \tau^2_{\Delta h_s} + \tau^2_{P}} \qquad (45)$$

7.5.4 *Uncertainty of corrections*

For the determination of the measuring uncertainty of results which have to be corrected, the correction curves of the manufacturer shall be recognized as correct in the tolerable range of correction.

If a correction is necessary beyond the acceptable range specified in 3.8.2 and 6.4, an additional allowance (e.g. one-third of the additional correction value which results from the excess of the variable beyond the acceptable correction range) may be agreed by the parties to avoid rejection of the test, or the correction value may be recalculated for the specific cycle.

The uncertainty of the correction variables (pressure, temperature, etc.) causes an uncertainty of the correction which in many cases is small enough to be neglected for the determination of the uncertainty of the result.

7.5.5 *Guiding values for the measuring uncertainty of results*

Establishing the measuring uncertainty of results for large and complicated installations generally calls for a considerable amount of calculating work which, in most cases, can only be carried out on the basis of simplified assumptions.

It is permissible to omit detailed calculation of the uncertainty value of the test result, if the parties to the test so agree and if it is clear from the results that the contract requirements have been fulfilled. This latter condition is satisfied if the test result, after allowing for a reliable estimate of the uncertainty value, still meets the guarantee.

Guiding values for the uncertainty of test results are given in Table III.

These guiding values, according to general experience, are an indication of the magnitude of measuring uncertainties for correct acceptance tests made in accordance with the requirements of Table III.

TABLE III

Guiding values for the uncertainty of test results

Reference line	Type of turbine	Result	For variables	Instrumentation	Relative measuring uncertainty for the result*
A	Back pressure turbine	η_{td}	$P_1 \cdot t_1 \cdot P_4$ P_b $\dot{m}_1$	As references 1, 2, 3, 5 As references 15, 16 As references 12, 13 Table II	$\Delta h_s > 400$ kJ/kg ± 1.5–2.0
B	Back pressure extraction turbine	η_{td}	As reference A in this table. However, P_e as reference 2, 3 in Table II and $\dot{m}_2$ as reference 12, in Table II		$\Delta h_s > 400$ kJ/kg ± 1.7–2.5
C	Condensing turbine	η_{td}	As reference A in this table. However, $\dot{m}_1$ from $\dot{m}_5$, t_{wi} and t_{wo} as references 10 and 11 in Table II		± 1.0–1.7
D	Condensing extraction turbine	η_{td}	As references B and C in this table		± 1.3–2.0
E	Condensing turbine with reheating	η_t	P_1, P_3, P_4 t_1, t_3, t_{11}, t_5 t_{wi}, t_{wo} etc. P_b $\dot{m}_1$, $\dot{m}_3$	As references 1, 2, 3 and 5, 6 As references 9, 10 and 11 As references 10, 11 As references 15, 16 in Table II Calculated from measurements with checked pressure difference devices (preferably in water part of the circuit) as reference 12 in Table II	± 0.9–1.2
F	Condensing turbine with saturated steam as initial steam	η_t	P_1 h_1 P_b	As references 1, 2, 3 in Table II Enthalpy according to special method As references 15, 16 in Table II All other measuring values as reference E in this table	± 1.1–1.6

* These values can be reduced mainly by increasing the accuracy of the determination of the primary flows.

APPENDIX A

FEEDWATER HEATER LEAKAGE AND CONDENSER LEAKAGE TESTS

(see 3.4.5)

A1. *Feedwater heater leakage tests*

Heaters may be checked for leaks, with the turbine out of service, by using the condensate or boiler feed pumps to maintain pressure on the water side of the heaters. Leakage is indicated by the accumulation of water in the hot well or shell of the heater and, if normal operating water pressure is maintained, the rate of leakage may be estimated. It should be realized, however, that the rate of leakage may vary with the heater temperature. The measured rate of leakage should be regarded only as an indication of the relative tightness of the heater and should not be used as the basis for correction of the primary flow measurement.

In some installations, it may be possible to check for heater leakage with the turbine in service if a tight shut-off of bled steam can be obtained and if there is no cascading of heater drains into the heater being checked.

If leakage is suspected, a useful method to check for heater leakage in actual service, is to inject a small quantity of a water treatment chemical into the condensate line before the feed heater. A check of the conductivity of the heater drain condensate from any suspected heater will show, in the case of a leak, a sudden rise in conductivity as the chemical passes through.

A2. *Condenser leakage tests*

Prior to and directly after the turbine tests, a hydraulic test should be made on the condenser by filling the steam space with water up to at least 20 cm above the top row of tubes and noting the leakage of water, if any, into the water boxes at inlet and outlet.

An alternative method of condenser test before and after the turbine tests is to place the condenser and turbine into full vacuum by closing off all access of air and steam, and delivering normal water supply to the condenser tubes with the cooling water pumps. Weighed leakage into the hot well is a measure of condenser leakage. The tightness of the condenser, immediately before and during turbine tests, should be checked by an electrical conductivity test of a sample of condensate from the condenser's hot well and a sample of the cooling water after it has been diluted by a known amount of distilled water.

Alternatively, the leakage can be checked by a chemical method or by a fluorescence method.

APPENDIX B

THROAT TAP NOZZLE
(see 4.3.2.1)

B1. *Design and manufacture*

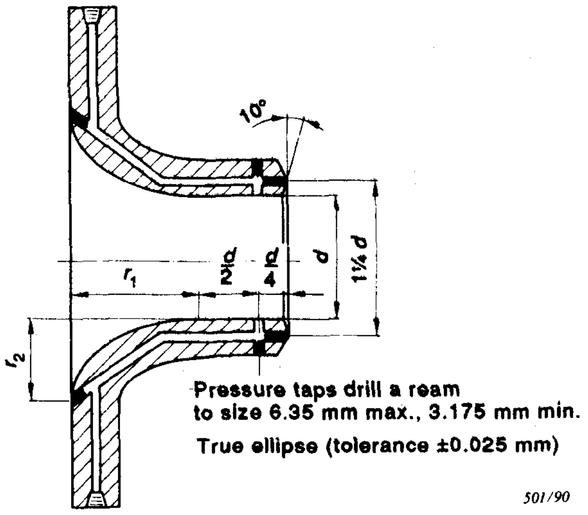
Because of the high degree of accuracy necessary, the following requirements are given regarding the design and manufacture of throat-tap nozzles for primary-flow measurement. Figure B1 is an example of a long-radius, low-ratio, nozzle shape with throat taps that satisfies these requirements.

a) The entrance shall be designed to give a favourable pressure gradient so that the boundary layer will be very thin in the throat section and there will be no flow separation. Also, the entrance shall be designed to give a uniform flow approaching the throat section. The cylindrical section of the nozzle shall have parallel walls and be concentric with the pipe (see B2*b)*). Any divergence may cause a peculiarly shaped coefficient of discharge versus the Reynolds number curve. However, slight convergence is acceptable, but no more than one thousandth of a mm per mm of throat length. The area in the plane of the throat taps shall be used in the coefficient calculation. Referring to Figure B1, the throat of the nozzle shall be round within the limits of ± 0.0002d as determined by measurements on at least four diameters in the plane of the throat taps. The nozzle shall be made from a corrosion-resistant material with known thermal-expansion coefficient and its surface should have a 10^{-4} mm finish or better, and shall be free from all burrs, scratches, imperfections, or ripples.

b) The pressure taps shall be at least two pressure-tap diameters deep. They shall be machined perpendicular to the bore surface, shall have sharp corners, and be free from burrs. The downstream pressure taps shall be machined in the throat of the nozzle in order to decrease the effect of downstream disturbances on this pressure measurement. The upstream taps shall be carefully made and shall be located one pipe diameter upstream from the nozzle entrance.

c) The final determination of the compliance with the above requirements is the shape of the coefficient of discharge versus the Reynolds number curve (see Figure B2) which should be determined for each set of taps to be used. The nozzle shall be used only in the range where the co-efficient of discharge is nearly constant.

d) In order to obtain maximum reading accuracy, the nozzle-throat diameter shall be selected to give the maximum deflection practicable, considering both the available pumping head and the manometer range. The manometer range should be selected to allow for fluctuations and for maximum flow which may be encountered. The nozzle shall not be used to measure flows which give manometer deflections of less than 1000 times the reading error or 150 mm of mercury, whichever is larger. When it is necessary to measure flow over a larger range than can be obtained by complying with this requirement, it is permissible to use additional nozzles with different throat diameters. These nozzles shall be sized so that one of the test points can be measured with either nozzle.


10°
r_1
r_2
$\frac{d}{2}$
$\frac{d}{4}$
d
$1\frac{1}{4}d$
Pressure taps drill a ream to size 6.35 mm max., 3.175 mm min.
True ellipse (tolerance ±0.025 mm)
501/90

Notes:

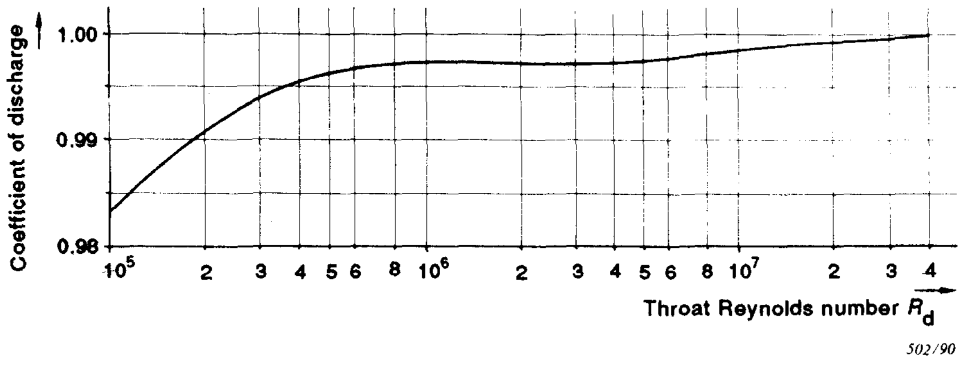
1. — Pressure tap holes to have square and sharp corners and to be free of burrs. The pressure tap holes are to be drilled and reamed before the final boring and polishing of the throat. A plug with a press fit is then inserted in the hole. The final boring and polishing operation should be done after the insertion of the plug. The plug should be made with provision for pulling it out of the hole after the polishing and machining is completed. After removal of this plug, any slight burr which might be left on the edge of the hole may be removed by taking a tapered piece of maple and rolling it around the pressure tap.

2. — The throat of the nozzle may converge as much as 0.001 mm per mm. No divergence is acceptable.

3. — Material to be corrosion resistant.

δ = pressure tap diameter

FIG. B1. — Throat-tap flow nozzle.



Throat Reynolds No. × 10^{-6}	C
	$\beta = 0.43$
0.1	0.9834
0.2	0.9909
0.3	0.9939
0.4	0.9953
0.5	0.9961
0.6	0.9966
0.8	0.9970
1.0	0.9972
2.0	0.9970
3.0	0.9970
4.0	0.9972
5.0	0.9974
6.0	0.9978
8.0	0.9980
10.0	0.9983
20.0	0.9991
30.0	0.9994
40.0	0.9998

FIG. B2. — Typical nozzle calibration curve for $\beta = 0.43$.
For other values of β within the range 0.25–0.50:
$C = C_{(\beta = 0.43)} + 0.011\,339\,\beta - 0.0049$.

B2. *Flow section*

a) The flow nozzle shall be installed in a flow section as shown in Figure B3. This flow section shall contain a flow straightener which divides the pipe cross section into at least 50 approximately equal spaces or a perforated multiplate flow straightener having approximately 200 holes per plate and there should be 20 diameters of straight run pipe upstream of the flow nozzle to ensure a sufficiently uniform velocity profile in the approaching stream. The requirement of at least 10 pipe diameters of straight pipe on the discharge side of the flow nozzle, of the same nominal size as the upstream pipe, ensures a reliable throat-pressure measurement.

b) The flow nozzle shall be centred in the pipe within 0.8 mm of the pipe axis. The pipe on either side of the flow nozzle shall be smooth, free from rust, scale and blisters, and the inside diameter measured at four points at any cross section shall not differ by more than 1 %. The upstream pipe section shall be bored as shown in Figure B4 or for the entire length of the inlet section.

c) The pipe joints at the flow nozzle shall have the inner bores square with the faces of the flanges. The compressed thickness of the gaskets shall not exceed 1.6 mm and the gaskets shall not extend within the pipe.

d) In order to reduce the possibility of thermal distortion of the nozzle, it is desirable that the pipe and the flanges of the flow section be made of a corrosion-resistant material having the same coefficient of expansion as the nozzle.

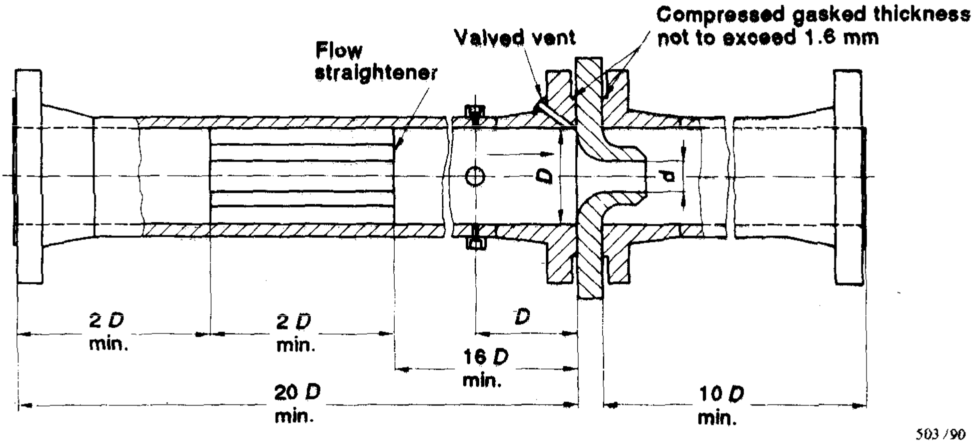


FIG. B3. — Flow section.

Note. — No obstruction, such as thermocouple, wells, backing rings, etc.

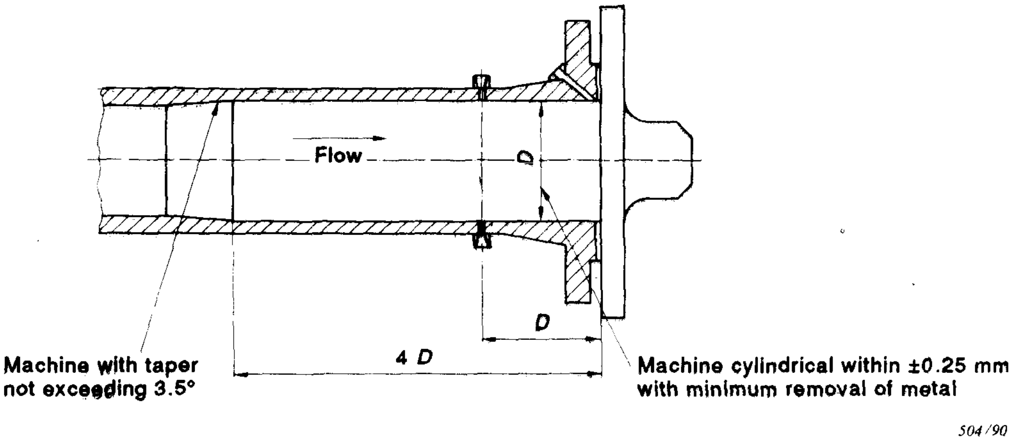

Flow
D
D
4 D
Machine with taper not exceeding 3.5°
Machine cylindrical within ±0.25 mm with minimum removal of metal
504/90

FIG. B4. — Boring in flow section upstream of nozzle.

B3. *Calibration*

a) Experience shows that the flow coefficient cannot be predicted within 0.1% and, therefore, it is necessary to calibrate the flow sections (see Figure B3). This calibration shall be undertaken only by recognized authorities under conditions similar to those in the actual installation. It is desirable that the physical construction of the piping in the calibrating set-up should be similar to that in the test setup from the standpoint of pipe configuration, immediately upstream and downstream of the flow-measuring section. Also, the Reynolds number, water temperature and other flow conditions should be as close to the test conditions as possible. If the calibration of the flow section does not comply with B3*b)*, the nozzle shall be carefully inspected to conform to the requirement of B1*a)*, corrected, if necessary, and the flow section recalibrated. If the recalibration still does not comply with B3*b)*, the flow section should be calibrated using different facilities, if available.

b) Calibration shall be conducted on at least two sets of pressure taps 180° apart. For each set of taps, the calibration curve (not necessarily each individual point) shall be within 0.25% of the reference curve and shall have the same slope (see reference curve Figure B2). When it is not practicable to calibrate at the test Reynolds number, the calibration Reynolds number should be obtained in accordance with B3*c)*. The extrapolation to higher Reynolds numbers will then be parallel to the reference curve. The end point of the extrapolation should not differ by more than 0.25% from the calibration curve coefficient at the highest Reynolds number attained during calibration.

The location of the primary-flow section in the cycle, its configuration, and the technique which is employed to obtain the flow measurements are critical and are discussed in subsequent paragraphs.

c) Below a throat Reynolds number of 2×10^6, there is a transition from laminar to turbulent boundary layer. This transition region should be established during calibration and shall be avoided during the test. It is permissible to extrapolate the calibration curve when the nozzle is used beyond the range of the calibrating facility provided the level of the coefficient is established at Reynolds numbers higher than the transition region. This extrapolation should be parallel to the curve shown in Figure B2 and subject to the restrictions of B3*b)*.

d) It is preferable to install flow sections immediately before the test. A slight iron-oxide film on the nozzle surface will usually collect during the test. If this film is very thin (less than 0.025 mm) and uniformly deposited, its effect on the accuracy of the flow measurement will be negligible. If the thickness of the deposit exceeds this value, or if the nature of the deposit is not uniform and the surface appears rough, either of two procedures may be followed:

- the nozzle may be cleaned, reinstalled, and the test repeated; or
- the flow-measuring section may be recalibrated.

Care has to be taken not to disturb the deposit before recalibration. If this calibration is significantly different from the calibration prior to the test, it is necessary that another set of runs be made under deposit-free conditions. The test results cannot be adjusted, since it is usually impossible to determine when the deposit formed on the nozzle.

APPENDIX C

EVALUATION OF MULTIPLE MEASUREMENTS, COMPATIBILITY
(see 5.2.3.2)

If several individual values for the same variable have been measured or determined (see 5.2.3.3) and their uncertainties have been evaluated, there is a good method for checking whether each of these several values is compatible with the weighted average value (see 5.2.3.2) and its uncertainty (see 7.1). It assumes that the uncertainties of the individual values have been determined correctly and that the individual values differ only randomly from the true value.

With

x_i = individual value
V_{x_i} = uncertainty of x_i
$\tilde{x}$ = weighted average value of all individual values (see 5.2.3.2)
$V_{\tilde{x}}$ = uncertainty of $\tilde{x}$ (see 7.1)

the compatibility criterion

$$\varepsilon_i = 1 - \frac{(x_i - \tilde{x})^2}{V^2{}_{x_i} - V^2{}_{\tilde{x}}}$$

is calculated for each individual value x_i.

If $\varepsilon_i < 0$,

the individual value x_i in question differs, possibly because of a systematic error in an unacceptable manner from the true value. This is a warning signal to investigate the value x_i and its uncertainty value V_{x_i} for undetected systematic errors.

The lower the value ε_i, the higher the probability of such an error.

If in the case of an incompatible value a physical inspection of the measuring installation is not practicable or evidence of a systematic error exists, this value should be eliminated from further computation.

If for all individual values

$$\varepsilon_i \geqslant 0$$

the statistical compatibility of the measured values x_i and their uncertainty V_{x_i} can be assumed.

APPENDIX D

MASS FLOW BALANCES

(see 5.2.3.3)

In cases where several primary mass flow measurements are carried out in the main flow of the cycle, a particular primary mass flow of the cycle (e.g. the initial steam flow) can be determined with the aid of mass flows balances by several largely independent measurements. Mass flow balances are not altogether independent when common mass flows occur in such balances. This can be taken into account for the determination of the average value and its uncertainty.

To set up the equations for mass flow balances, it is further necessary to measure certain secondary mass flows as well as the level changes of tanks and other storage vessels in the cycle.

The unaccounted-for cycle loss is determined from:

- the changes ΔI of all storage volumes, divided by the duration z of the test,
- the measured losses and
- the make-up water flow (if any).

This loss is distributed to the different parts of the plant according to general convention and special observations and is considered accordingly in the mass flow balances.

Level changes which are not attributable to losses shall be added as mass flows to the respective mass flow values.

Figure D1 illustrates the cycle diagram of a single-reheat steam turbine plant. Four primary mass flow measurements are made while four secondary mass flows and the changes in the contents of two tanks are measured.

The initial steam flow can be calculated from the balances in the following four ways:

a) $\dot{m}_{1,\mathrm{I}} = \dot{m}_1 - \dot{m}_{\mathrm{V}2}$

b) $\dot{m}_{1,\mathrm{II}} = \dot{m}_{\mathrm{K}} \pm \dot{m}_{\Delta \mathrm{IDA}} + \dot{m}_{\mathrm{H}5} + \dot{m}_{\mathrm{H}4} + \dot{m}_{\mathrm{H}3} - \dot{m}_{\mathrm{V}2} - \dot{m}_{\mathrm{V}1}$

c) $\dot{m}_{1,\mathrm{III}} = \dot{m}_{10} - \dot{m}_{\mathrm{V}2} - \dot{m}_{\mathrm{V}1}$

d) $\dot{m}_{1,\mathrm{IV}} = \dot{m}_{11} + \dot{m}_{\mathrm{is}} - \dot{m}_{\mathrm{V}2} - \dot{m}_{\mathrm{V}1}$

with distribution on $\dot{m}_{\mathrm{V}1}$ to $\dot{m}_{\mathrm{V}3}$ according to convention and $\sum_i \dot{m}_{\mathrm{V}_i} = \frac{\Delta I}{z}$ a duration time z of the acceptance test run.

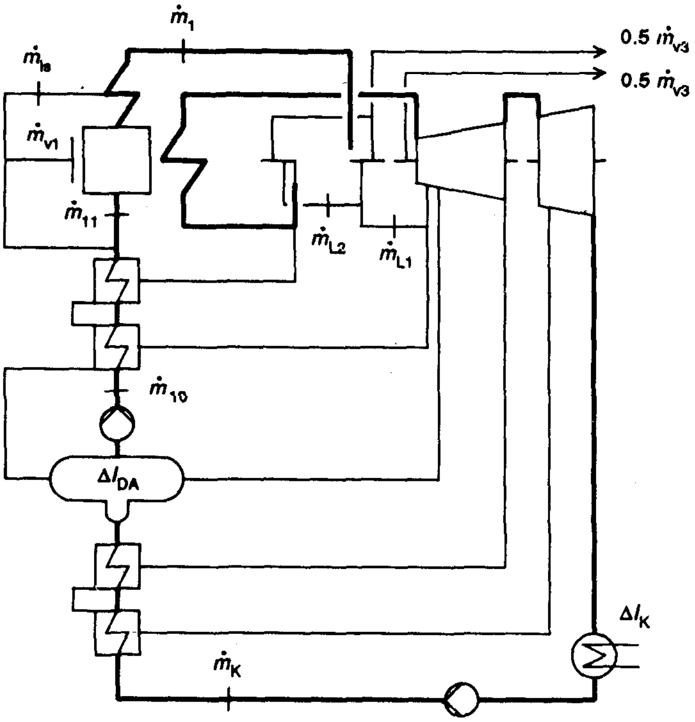


FIG. D1. — Diagram of cycle for plant with steam turbine with single reheating and five stages of regenerative feedwater heating extraction.

The various values $\dot{m}_j$ established from the mass flow balances with their measuring uncertainty $V_{\dot{m}_j}$ shall be checked for compatibility (see 5.2.3.2).

The weighted average value $\bar{\dot{m}}_l$ according to equation (24) is determined from the compatible values.

Individual mass flows (e.g. $\dot{m}_K$ or $\dot{m}_3$) required for further evaluation shall be calculated with the aid of the measured or calculated secondary flows from the averaged initial steam flow $\dot{m}_l$.

If a mass flow equation which is to be used for that calculation has led to an incompatible initial steam flow value in the mass flow balance evaluation, the measured mass flow contained in it should be checked.

For each mass flow balance a difference $\dot{m}_n$ can be calculated:

$$\dot{m}_n = \bar{\dot{m}}_l - \dot{m}_{l,n}$$

If the uncertainty of the mass flow balance value $\dot{m}_{l,n}$ is V_n and the uncertainty of the mass flow $\dot{m}_j$ is V_j a mass flow balance correction

$$\dot{m}_{B,j,n} = \frac{\dot{m}_n V^2_j}{V^2_n}$$

can be calculated.

The corrected mass flow is then:

$$\dot{m}_{j,corr} = \dot{m}_j \pm \dot{m}_{B,j,n}$$

The mass flow balance corrections $\dot{m}_{B,j}$ for several balance equations will deviate but very slightly and can be averaged arithmetically.

APPENDIX E

TYPICAL GENERALIZED CORRECTION CURVES FOR CORRECTION OF TEST RESULTS ACCORDING TO GUARANTEE CONDITIONS

(see 6.6.4)

General

The curves E1 to E17 may be used to correct the measured value of thermal efficiency for the influence of deviations of the operating parameters from the specified values during the test. They give correction factors E according to the definition in 6.5 and can be used for the correction procedure according to 6.6*a)* only.

The curves are prepared for single point guarantees and they therefore also include data for computation of deviations of measured main steam flow from guaranteed flow (E1 to E3). In case of guarantees and parameters given as functions of variable main steam flow (efficiency or heat rate curves), the measured main steam flow will be considered as guaranteed and the correction curves for main steam flow (E1 to E3) may therefore be ignored.

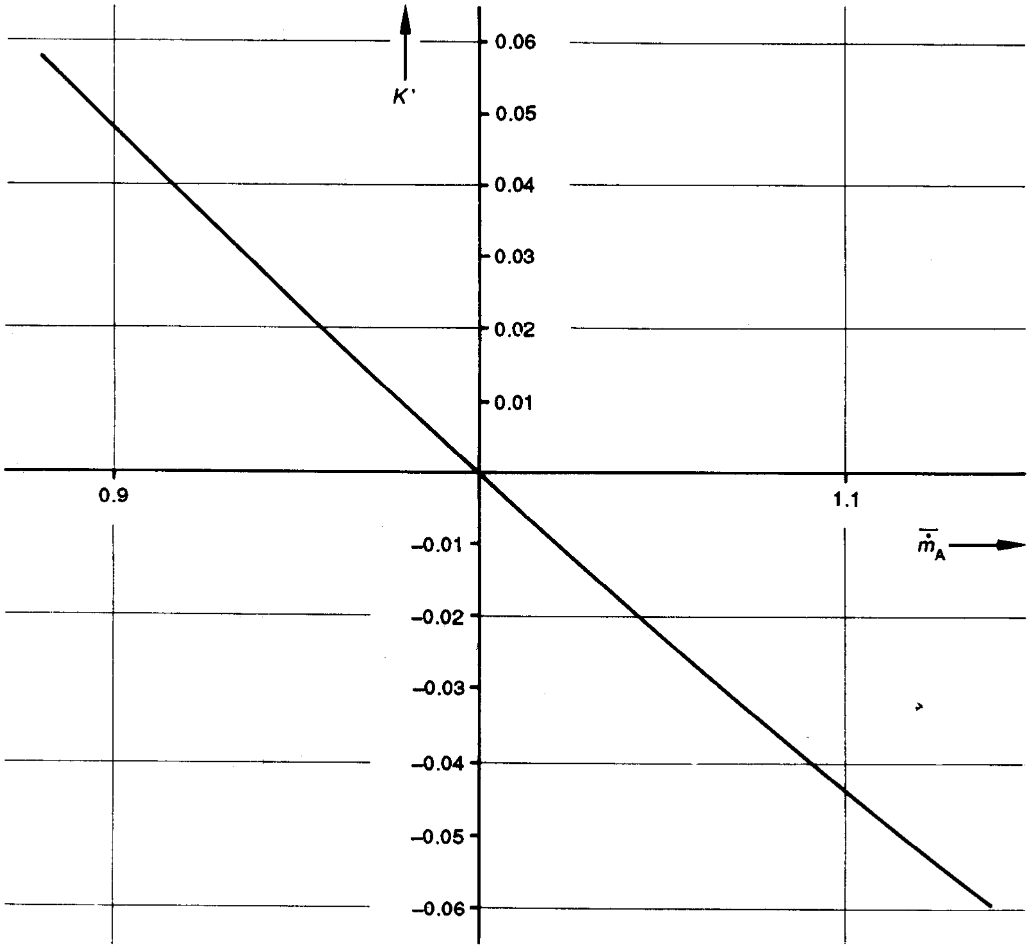
The application of certain curves is in some cases dependent on whether or not the parameters concerned are the subject of guarantee.

The curves may be used to correct for deviations of operating parameters etc., within the limits defined in 3.8.2 (Table I) and shall not be extrapolated.

The data are valid for condensing turbines of ratings higher than 30 MW and parameters which are commonly used in conventional and nuclear power stations.

The uncertainties of the corrections for the various parameters refer to probability $P = 0.95$ and are independent of each other and the resulting uncertainty may be calculated accordingly.

Mathematical formulations representing the curves in E1 to E17 are given in Table EI for programming purposes.



$$\eta_{t\cdot c} = \eta_{t\cdot m} \cdot F$$

$$F = 1 + K\left(\frac{T_v}{\Delta h_s}\right)_g$$

T_v = absolute temperature at exhaust pressure (K)

Δh_s = isentropic enthalpy drop, main steam/exhaust pressure (kJ/kg)

$\bar{\dot{m}}_A = \dfrac{\text{main steam flow as measured}}{\text{main steam flow as guaranteed}}$

K = K' for reheat and/or external moisture separation

K = 0.83 K' without reheat and without external moisture separation

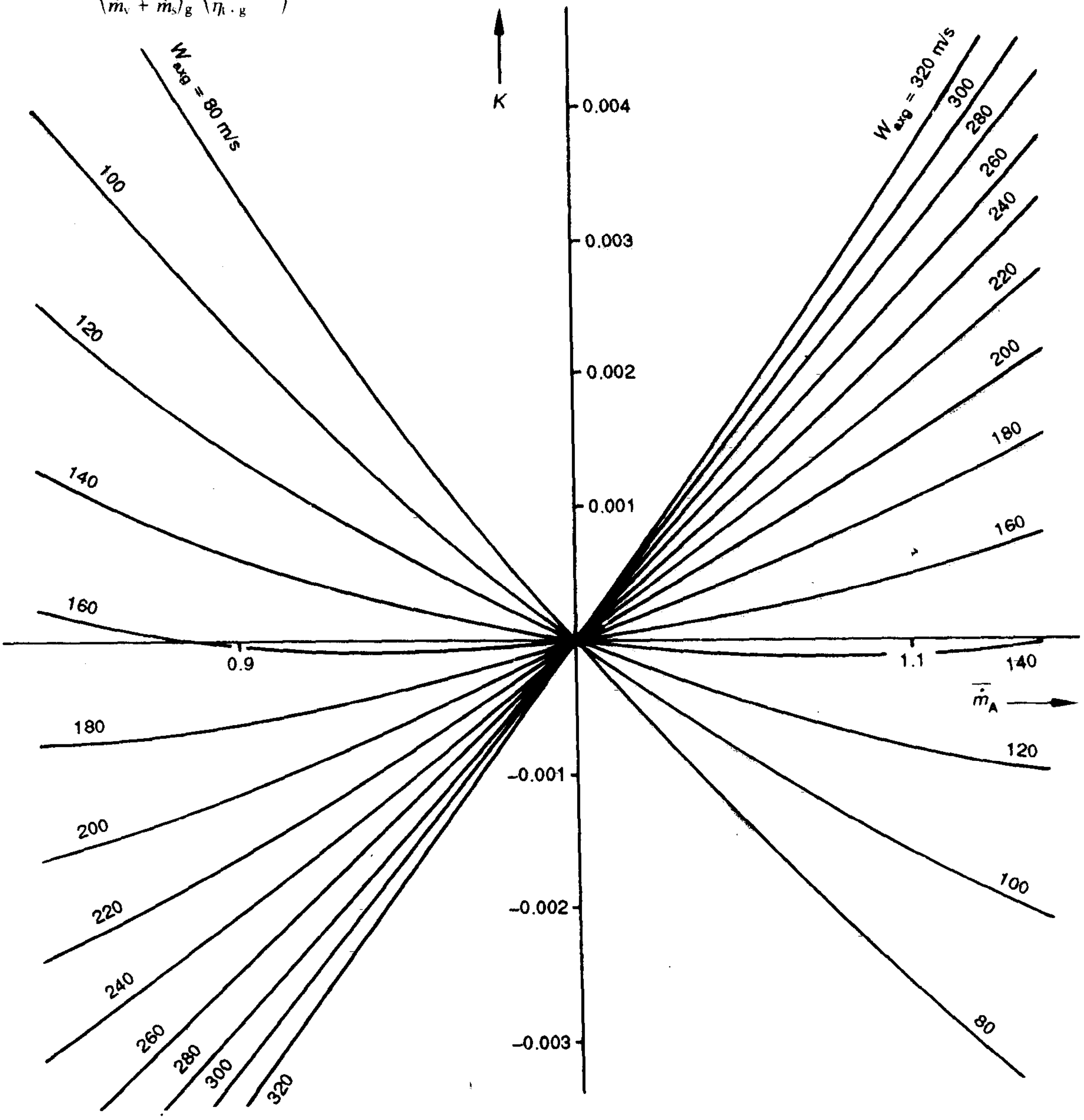
Valid for: Condensing turbines with regenerative feedwater heating, superheated or wet main steam with or without reheat, main steam enthalpy constant, output >0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.15 \mid F - 1 \mid$.

FIG. E1. — Correction curve for main flow influence on bowl pressure.

$$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$$

$$F = 1 + K \left(\frac{\dot{m}_v}{\dot{m}_v + \dot{m}_s} \right)_g \left(\frac{1}{\eta_{t \cdot g}} - 1 \right)$$



$\dot{m}_v$ = exhaust flows of main turbine (kg/s)

$\dot{m}_s$ = exhaust flow of BFP - auxiliary turbine (kg/s)

$\bar{\dot{m}}_A = \dfrac{\text{main steam as measured}}{\text{main steam as guaranteed}}$

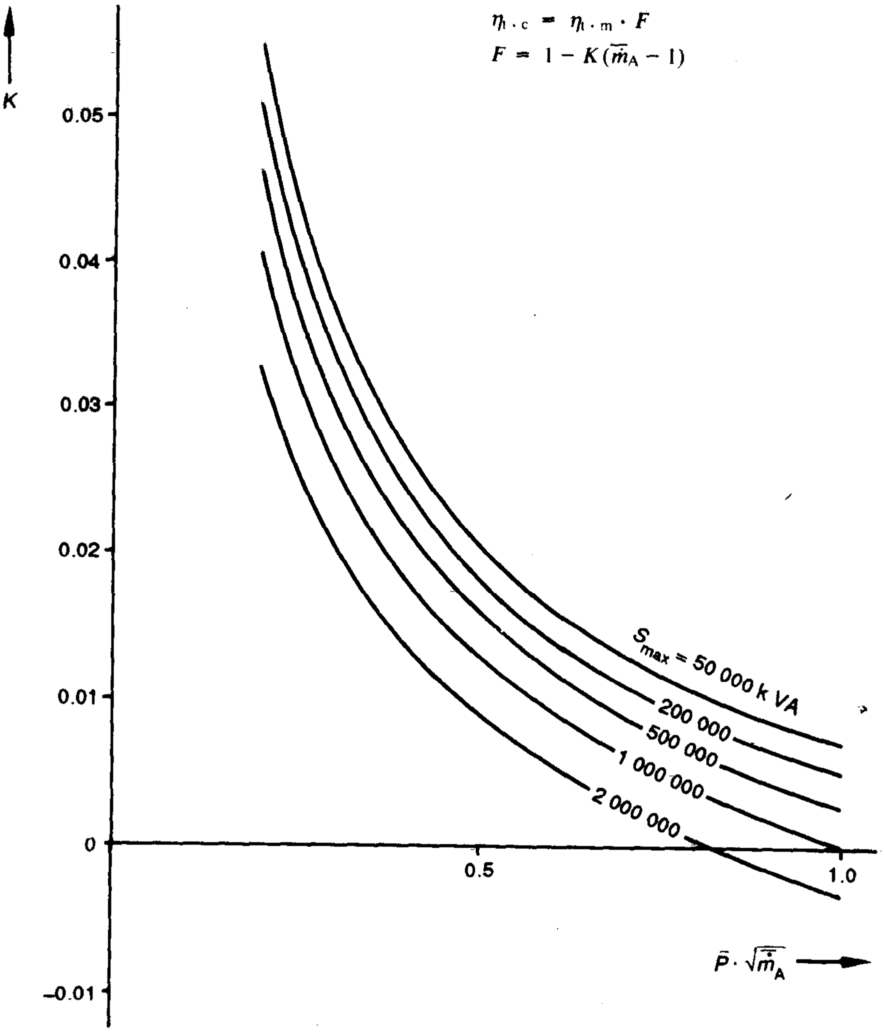
W_{axg} = axial outlet velocity as guaranteed (m/s)

u = peripheral speed of last stage wheel (pitch diameter) (m/s)

Valid for: Condensing turbines with regenerative feedwater heating, wet or superheated main steam, with or without reheat, exhaust pressure < 0.3 bar, $u = 200$ to 450 m/s, output ≥ 0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.2 \mid F - 1 \mid$.

FIG. E2. — Correction curve for main steam flow influence on exhaust losses.



$\overline{\dot{m}}_A = \dfrac{\text{main steam flow as measured}}{\text{main steam flow as guaranteed}}$

$\bar{P} = \dfrac{\text{generator output as guaranteed}}{\text{max. generator output}}$

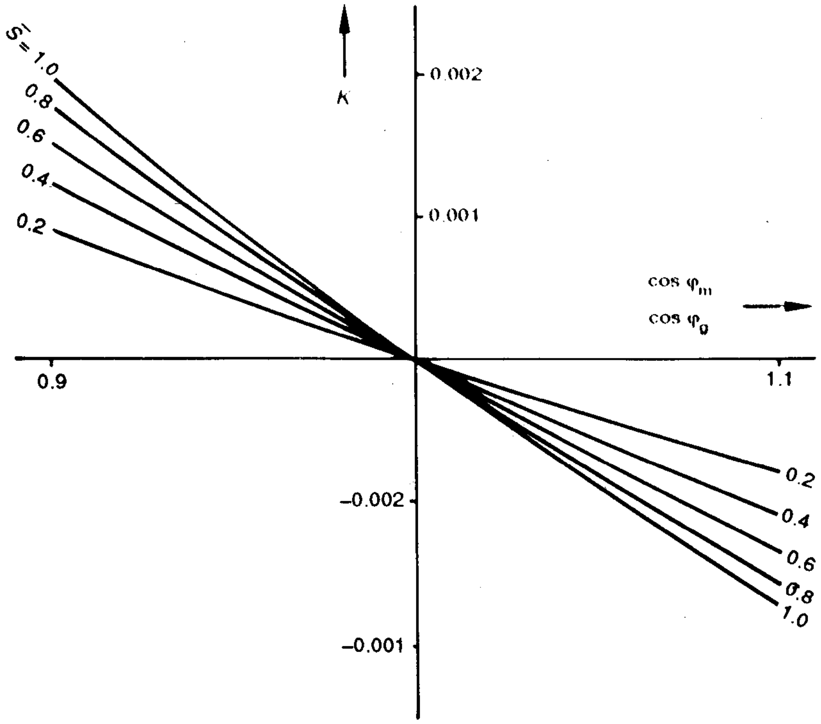
S_{max} = max. apparent generator output (kVA)

Valid for: Two-pole and four-pole generators, cooling systems adequate to S_{max}, $S_{max} > 30\,000$ kVA.

Uncertainty of computation: $\Delta F = \pm(|0.3\ K| + 0.003) \cdot |\overline{\dot{m}}_A - 1|$.

FIG. E3. — Correction curve for main steam flow influence on mechanical efficiency and generator efficiency.

$$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$$
$$F = 1 + K$$

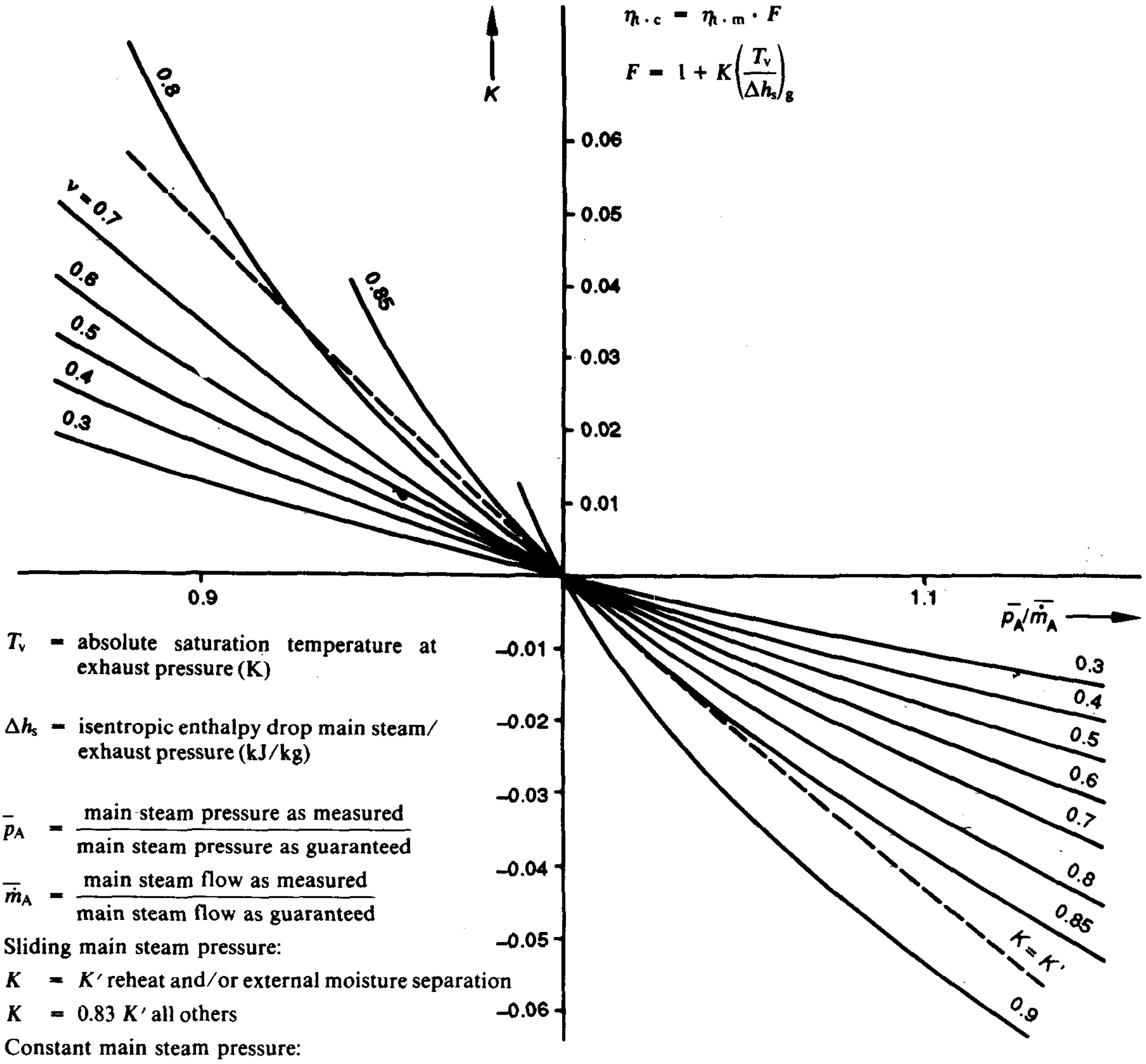


$\cos \varphi$ = load factor
S = apparent generator output (kVA)
S_{max} = max. apparent generator output (kVA)
$\bar{S} = \left(\dfrac{S}{S_{max}}\right)_g$

Valid for: $S_{max} > 30$ MVA, $\cos \varphi = 0.7$ to 1.0, active output = constant.

Uncertainty of computation: $\Delta F = \pm 0.4 \mid F - 1 \mid$.

FIG. E4. — Correction curve for load factor.



$$\eta_{t\cdot c} = \eta_{t\cdot m}\cdot F$$

$$F = 1 + K\left(\frac{T_v}{\Delta h_s}\right)_g$$

T_v = absolute saturation temperature at exhaust pressure (K)

Δh_s = isentropic enthalpy drop main steam/exhaust pressure (kJ/kg)

$\bar{p}_A = \dfrac{\text{main steam pressure as measured}}{\text{main steam pressure as guaranteed}}$

$\bar{\dot{m}}_A = \dfrac{\text{main steam flow as measured}}{\text{main steam flow as guaranteed}}$

Sliding main steam pressure:

K = K' reheat and/or external moisture separation

K = 0.83 K' all others

Constant main steam pressure:

Throttle control	$K = 0.11\,K'$ steam/steam reheat $K = 0$ all others
Nozzle control	$K = 0.11 + 0.89\,K(v)$ steam/steam reheat $K = 0.83\,K(v)$ without reheat and without external moisture separation $K = K(v)$ all others

$v = v_0^{\mu}$

$\mu = \dfrac{18\,(h_{Ag} - 1925)}{u^2}$ one-row governing stage

p_R = wheel chamber pressure (bar)

h_A = main steam enthalpy (kJ/kg)

$v_0 = \left(\dfrac{p_R}{p_A}\right)_g$

$\mu = \dfrac{8\,(h_{Ag} - 1925)}{u^2}$ two-row governing stage

p_A = main steam pressure (bar)

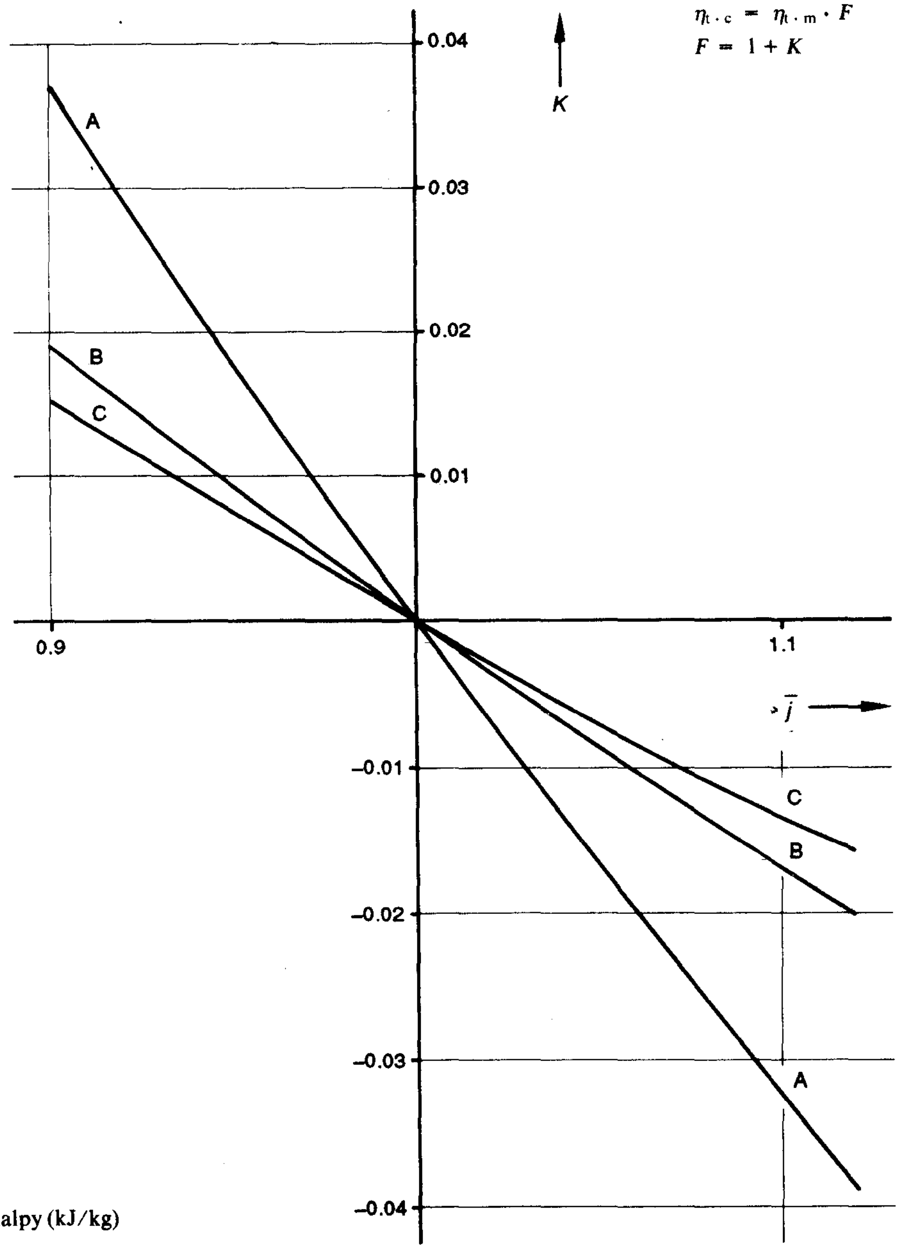
u = control stage peripherical speed pitch diameter (m/s)

Valid for: Condensing turbines with regenerative feedwater heating, wet or superheated main steam, with or without reheat, main steam enthalpy constant, $v \cdot \bar{\dot{m}}_A / \bar{p}_A < 0.91$, load >0.2 maximum load.

Uncertainty of computation: $\Delta F = \pm\left(0.15\,|\,F - 1\,| + 0.002\left|\dfrac{\bar{p}_A}{\bar{\dot{m}}_A} - 1\right|\right).$

FIG. E5. — Correction curve for main steam pressure.

This correction allows for deviation in *flow capacity* at agreed valve position (for deviation in *flow*, see F1).



$\bar{j} = \dfrac{h_m - 1925}{h_g - 1925}$

h = specific enthalpy (kJ/kg)

Curve A: Main steam of conventional turbines without reheat.

Curve B: Saturated (or nearly saturated) main steam of turbines with external moisture separation, without reheat.

Curve C: Main steam and reheat steam of conventional turbines with single reheat.
Saturated (or nearly saturated) main steam of turbines with external moisture separator and reheat.

Valid for: Condensing turbines with regenerative feedwater heating, output >0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm(0.15 \mid F-1 \mid + 0.002 \mid \bar{j}-1 \mid)$.

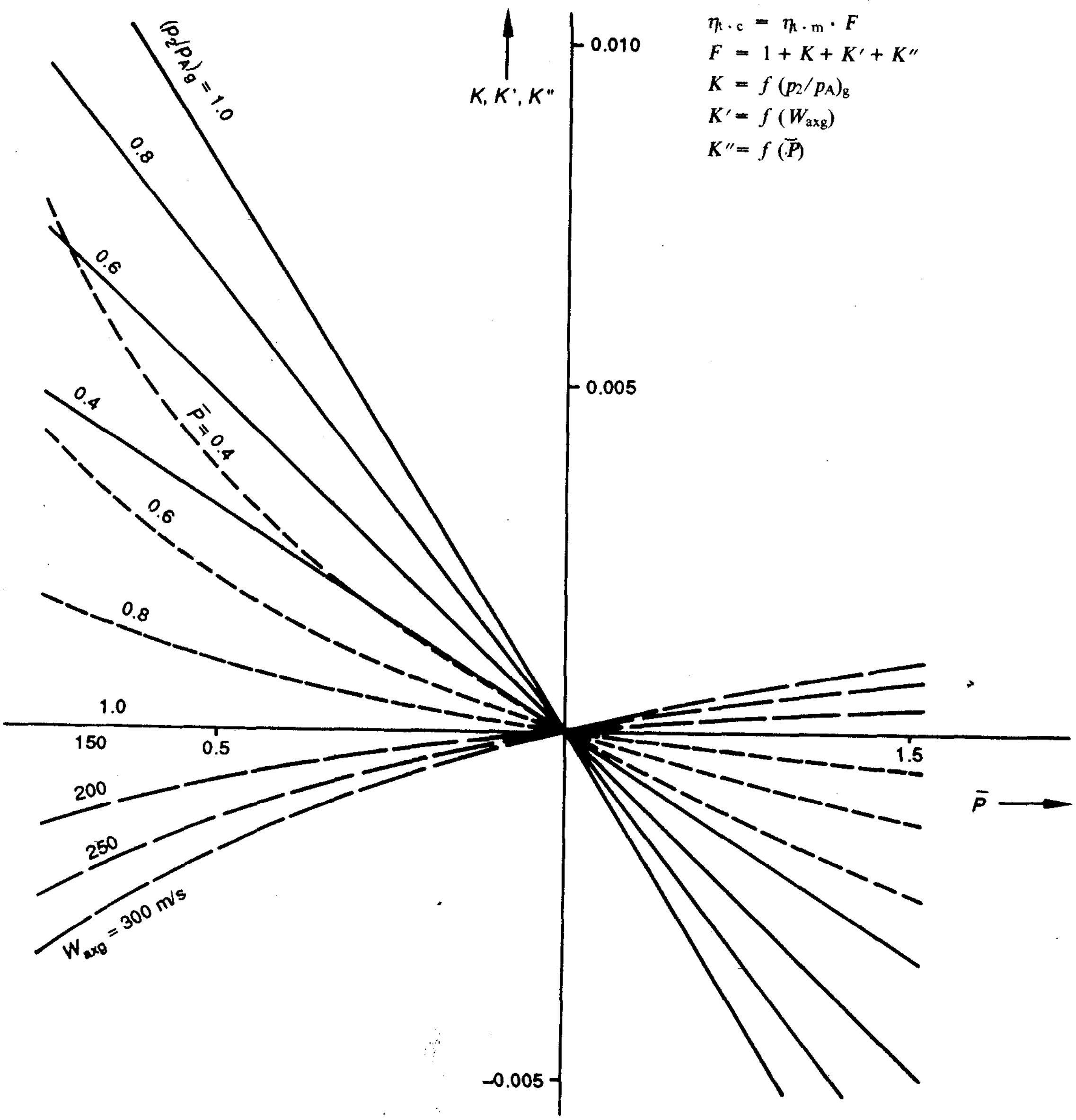
FIG. E6. — Correction curve for steam enthalpies.

$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$
$F = 1 + K$
$K = 0.21\,(x_g - x_m)$
x = quality, i.e., dryness fraction of wet steam by weight beyond moisture separator

Valid for: Condensing turbines with regenerative heating system, wet (or nearly wet) main steam, with or without following reheat.

Uncertainty of computation: $\Delta F = \pm 0.2 \mid F-1 \mid$.

FIG. E7. — Correction for final moisture of external separator.



$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$

$F = 1 + K + K' + K''$

$K = f\,(p_2/p_A)_g$

$K' = f\,(W_{axg})$

$K'' = f\,(\bar{P})$

$\bar{p} = \dfrac{p_{Ag} \cdot p_{2m}}{p_{2g} \cdot p_{Am}}$

p_2 = saturation pressure at ϑ_2 (bar)

ϑ_2 = reheat temperature (°C)

p_A = main steam pressure (bar)

$\bar{P} = \dfrac{\text{generator output as guaranteed}}{\text{max. generator output}}$

W_{axg} = last stage axial outlet velocity as guaranteed (m/s)

Valid for: Condensing turbines with regenerative feedwater heating, wet or slightly superheated main steam, final reheat by main steam, output >0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.2\sqrt{K^2 + K'^2 + K''^2}$.

FIG. E8. — Correction curve for reheater terminal difference.

Main steam/steam reheater stage only, pressure losses on the main steam side are included in this correction, see definition of $\bar{p}$.

If the condenser is included in the guarantee, $\bar{p}$ will not be computed on the basis of the measured exhaust pressure, but by means of the expected exhaust pressure for measured cooling water flow and cooling water temperatures (see E10).

$$\eta_{t\cdot c} = \eta_{t\cdot m} \cdot F$$

$$F = \frac{1}{1 - K\left(\dfrac{\dot{m}_v}{\dot{m}_v + \dot{m}_s}\right)_g \cdot \dfrac{u\,(0.185 + \sin\beta)}{10\,000}\left(\dfrac{1}{\eta_{t\cdot g}} - 1\right)}$$

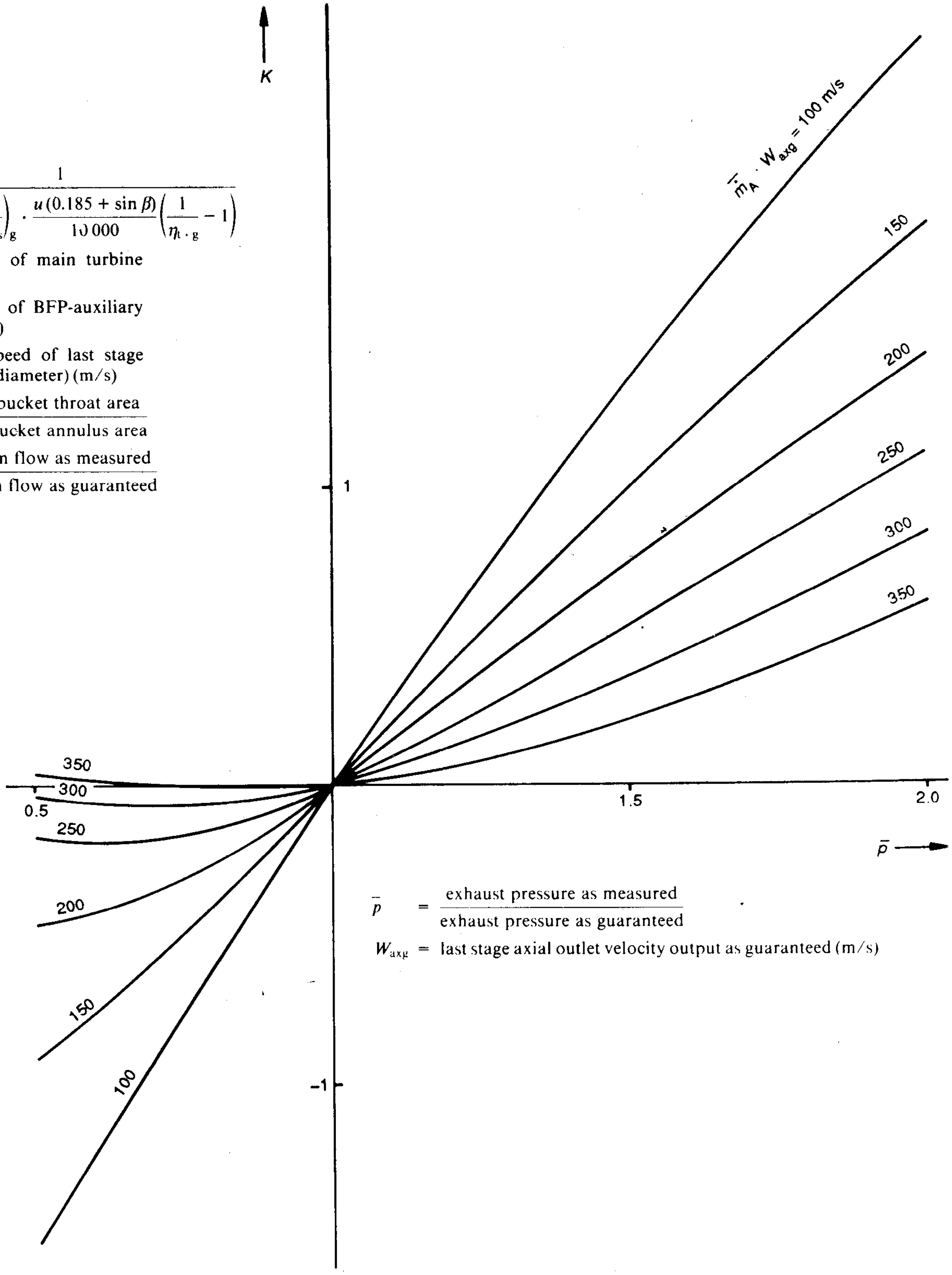
$\dot{m}_v$ = exhaust flow of main turbine (kg/s)

$\dot{m}_s$ = exhaust flow of BFP-auxiliary turbine (kg/s)

u = peripheral speed of last stage wheel (pitch diameter) (m/s)

$$\sin\beta = \frac{\text{last stage bucket throat area}}{\text{last stage bucket annulus area}}$$

$$\bar{\dot{m}}_A = \frac{\text{main steam flow as measured}}{\text{main steam flow as guaranteed}}$$



$$\bar{p} = \frac{\text{exhaust pressure as measured}}{\text{exhaust pressure as guaranteed}}$$

W_{axg} = last stage axial outlet velocity output as guaranteed (m/s)

Valid for: Condensing turbines with regenerative feedwater heating, with or without reheat, superheated or wet main steam, exhaust pressure < 0.3 bar, u = 200 m/s to 450 m/s, output > 0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm(0.15\,|\,F - 1\,| + 0.002\,|\,\bar{p} - 1\,|)$.

FIG. E9. — Correction curve for exhaust pressure.

If the condenser is included in the guarantee, $\bar{p}$ (see E9) will not be computed on the basis of the measured exhaust pressure, but by means of the following equation.

$$\bar{p} = \frac{p}{p_g}$$

$$\vartheta = \vartheta_{2m} + \frac{\vartheta_{2m} \cdot \vartheta_{1m}}{e^A - 1}$$

$$A = \left(\frac{\dot{m}_g}{\dot{m}_m}\right)^{0.5} \cdot \frac{1.4 - \dfrac{17}{\vartheta_{1m} + 20}}{1.4 - \dfrac{17}{\vartheta_{1g} + 20}} \cdot \ln\left(\frac{\vartheta_{2g} - \vartheta_{1g}}{\vartheta_g - \vartheta_{2g}} + 1\right)$$

ϑ = expected exhaust temperature (°C)

p = expected exhaust pressure (saturation pressure at ϑ) (bar)

ϑ_2 = cooling water exit temperature (°C)

ϑ_1 = cooling water inlet temperature (°C)

e = natural number (= 2.718)

ln = napierian logarithm

$\dot{m}$ = cooling water flow (kg/s)

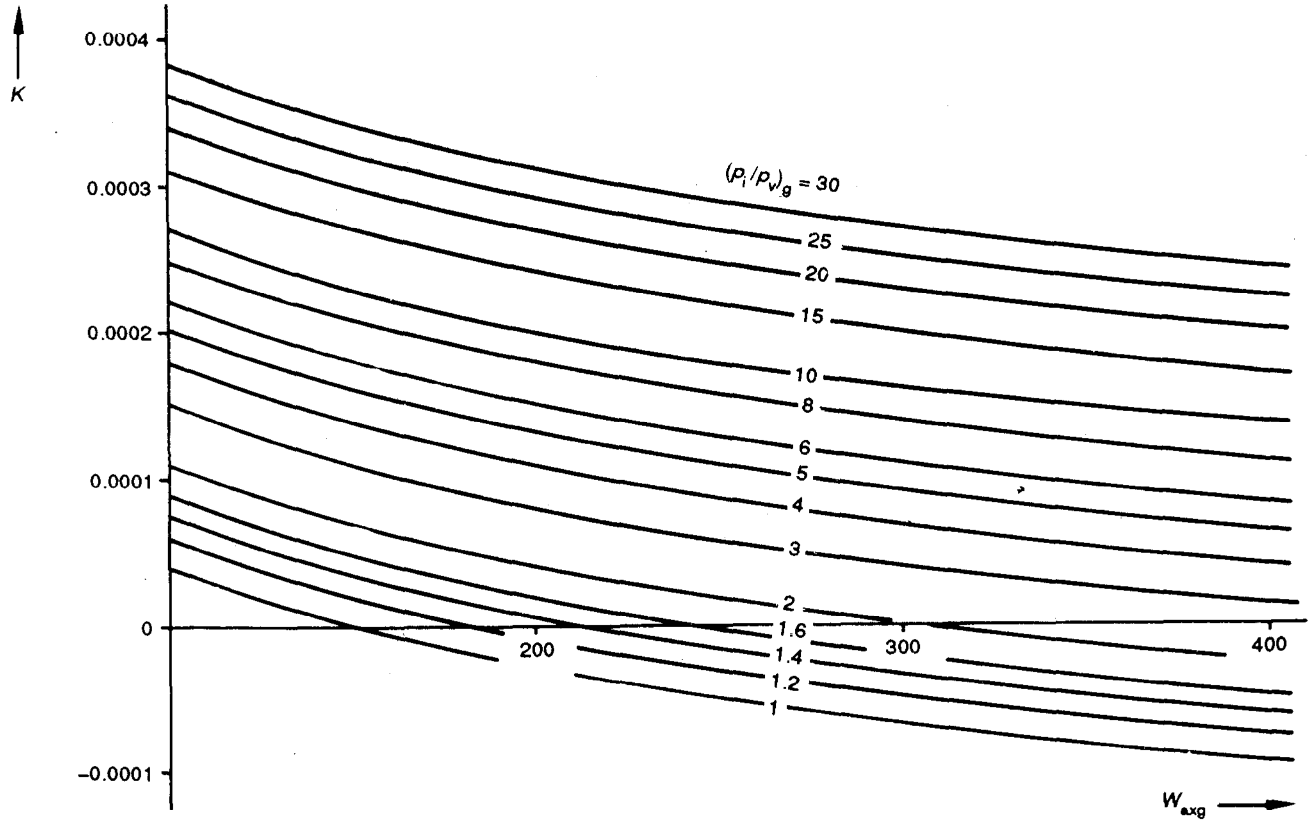
Valid for: Surface condenser of conventional design, cooling water velocity in tubes > 0.8 m/s.

Uncertainty of computation: $\Delta\bar{p} = \pm 0.1\,(\bar{p} - 1)$.

FIG. E10. — Correction for condenser cooling water flow and inlet temperature.

$$\eta_{i\cdot c} = \eta_{i\cdot m} \cdot F$$

$$F = 1 + K\left(\frac{1}{\eta_{i\cdot g}} - 1\right) \left[(\vartheta_v - \vartheta_w)_m - (\vartheta_v - \vartheta_w)_g\right]$$



p_v = exhaust pressure (bar)

ϑ_v = saturation temperature at exhaust pressure (°C)

ϑ_w = condensate temperature before lowest heater (°C)

p_l = pressure of lowest bleeding (bar)

W_{axg} = last stage annulus velocity as guaranteed (m/s)

Valid for: Condensing turbines with regenerative heating system, wet or superheated main steam, with or without reheat, exhaust pressure < 0.3 bar, output > 0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.00004 \cdot \left| (\vartheta_v - \vartheta_w)_m - (\vartheta_v - \vartheta_w)_g \right|$.

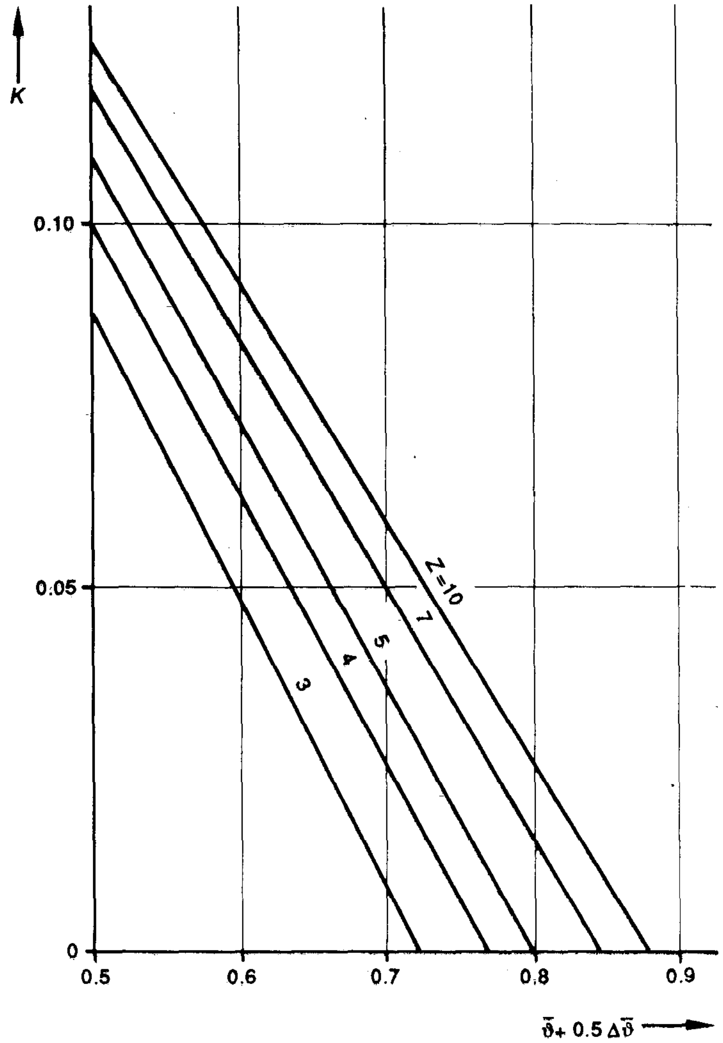
FIG. E11. — Correction curve for main condensate subcooling.

$$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$$

$$F = 1 - K\Delta\bar{\vartheta}$$

$$\Delta\bar{\vartheta} = \frac{\vartheta_{Em} - \vartheta_{Eg}}{\vartheta'_{Ag} - \vartheta_{Wg}}$$

$$\bar{\vartheta} = \frac{\vartheta_{Eg} - \vartheta_{Wg}}{\vartheta'_{Ag} - \vartheta_{Wg}}$$



ϑ_E = final feedwater temperature (°C)

ϑ'_A = saturation temperature for main steam pressure (°C), for $p_l > 22.1$ bar use $\vartheta'_A = 374$ °C

ϑ_W = condenser discharge temperature (°C)

Z = number of feed water heaters (–)

Remarks: Deviating terminal difference and extraction line pressure loss are to be corrected in addition, also for the highest heater.

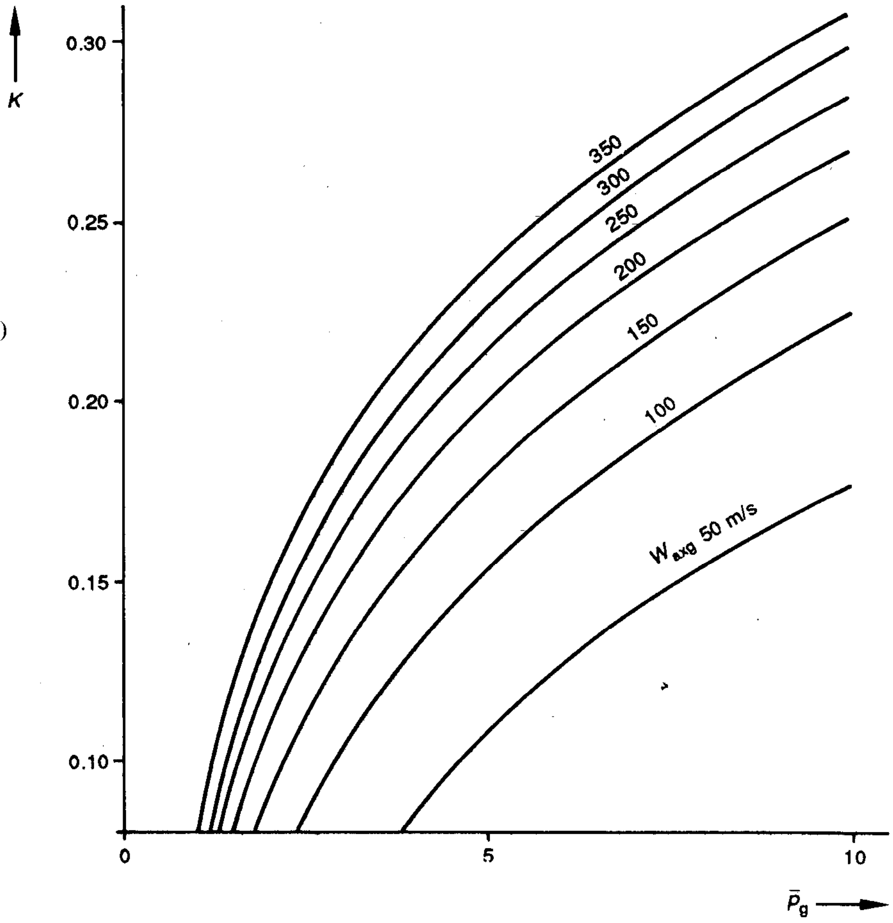
Valid for: Condensing turbines with regenerative feedwater heating ($\vartheta_E > 170$ °C, $Z > 2$), wet or superheated main steam, with or without reheat, output >0.2 maximum output, $|\Delta\bar{\vartheta}| < 0.05$.

Uncertainty of computation: $\Delta F = \pm 0.1\ (K + 0.1)\ \Delta\bar{\vartheta}$.

FIG. E12. — Correction curve for final feedwater temperature.

$\eta_{t\cdot c} = \eta_{t\cdot m} \cdot F$

$F = 1 + (K + \Delta K)\ (\overline{\dot{m}}_{m} - \overline{\dot{m}}_{g})$



$\bar{p}$ = $\hat{p}$ for sliding main steam pressure

$\bar{p}$ = $\hat{p}\,[1 - (\bar{P} - 1)^2 + 0.12\,(\bar{P} - 1)]$ for constant main steam pressure, nozzle control

$\bar{p}$ = $\hat{p} \cdot \bar{P}$ for constant main steam pressure throttle control

$\hat{p} = \dfrac{\text{main steam pressure}}{\text{HP exhaust pressure}}$

$\bar{P} = \dfrac{\text{generator output as guaranteed}}{\text{max. generator output}}$

ΔK = 0 for water extraction at feedpump

ΔK = 0.02 for water extraction behind highest heater

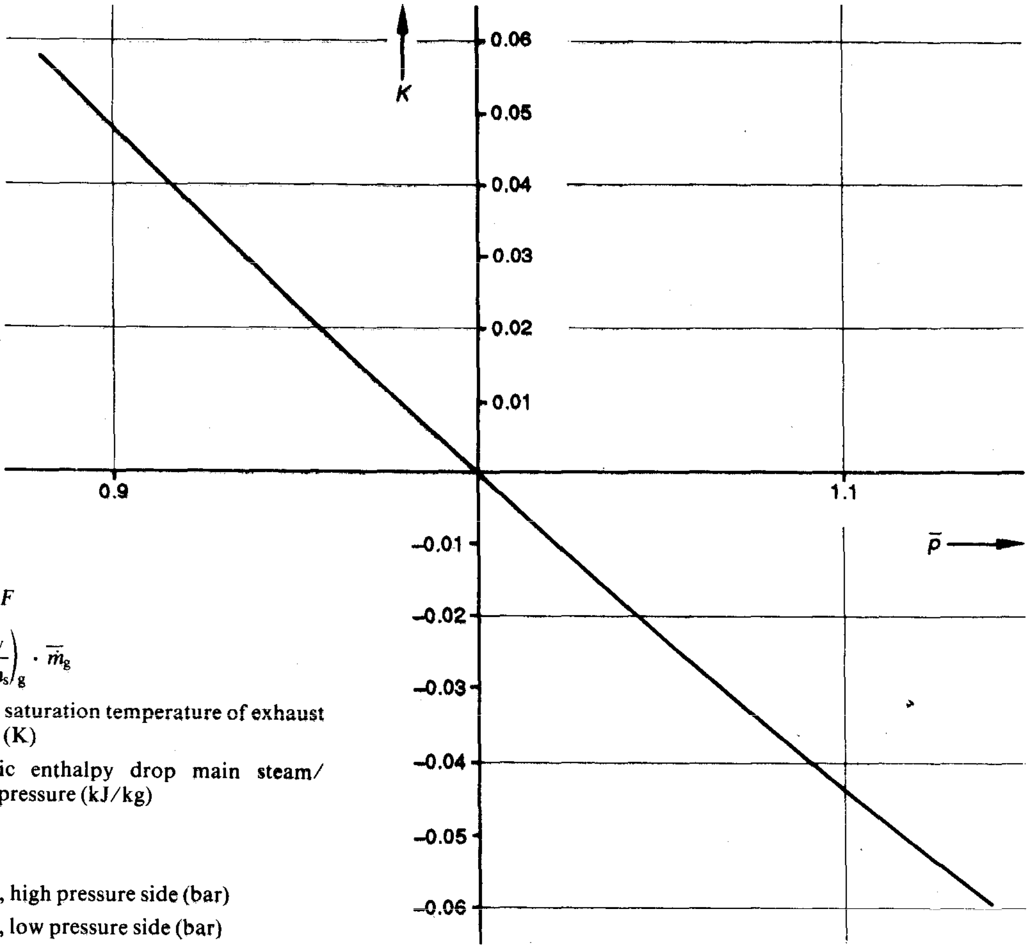
$\overline{\dot{m}} = \dfrac{\text{desuperheating water flow}}{\text{main steam flow}}$

W_{axg} = last stage axial outlet velocity as guaranteed (m/s)

Valid for: Condensing turbines with regenerative feedwater heating and single reheat, main steam and reheat temperatures >470 °C, output >0.2 maximum output reheat pressure sliding or constant.

Uncertainty of computation: $\Delta F = \pm\, 0.15\ |\ F - 1\ |$.

FIG. E13. — Correction curve for reheater desuperheating water.



$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$

$F = 1 + K \left(\frac{T_V}{\Delta h_s} \right)_g \cdot \bar{m}_g$

T_V = absolute saturation temperature of exhaust pressure (K)

Δh_s = isentropic enthalpy drop main steam/exhaust pressure (kJ/kg)

$\bar{p} = \frac{p_{1g} \cdot p_{2m}}{p_{2g} \cdot p_{1m}}$

p_1 = pressure, high pressure side (bar)

p_2 = pressure, low pressure side (bar)

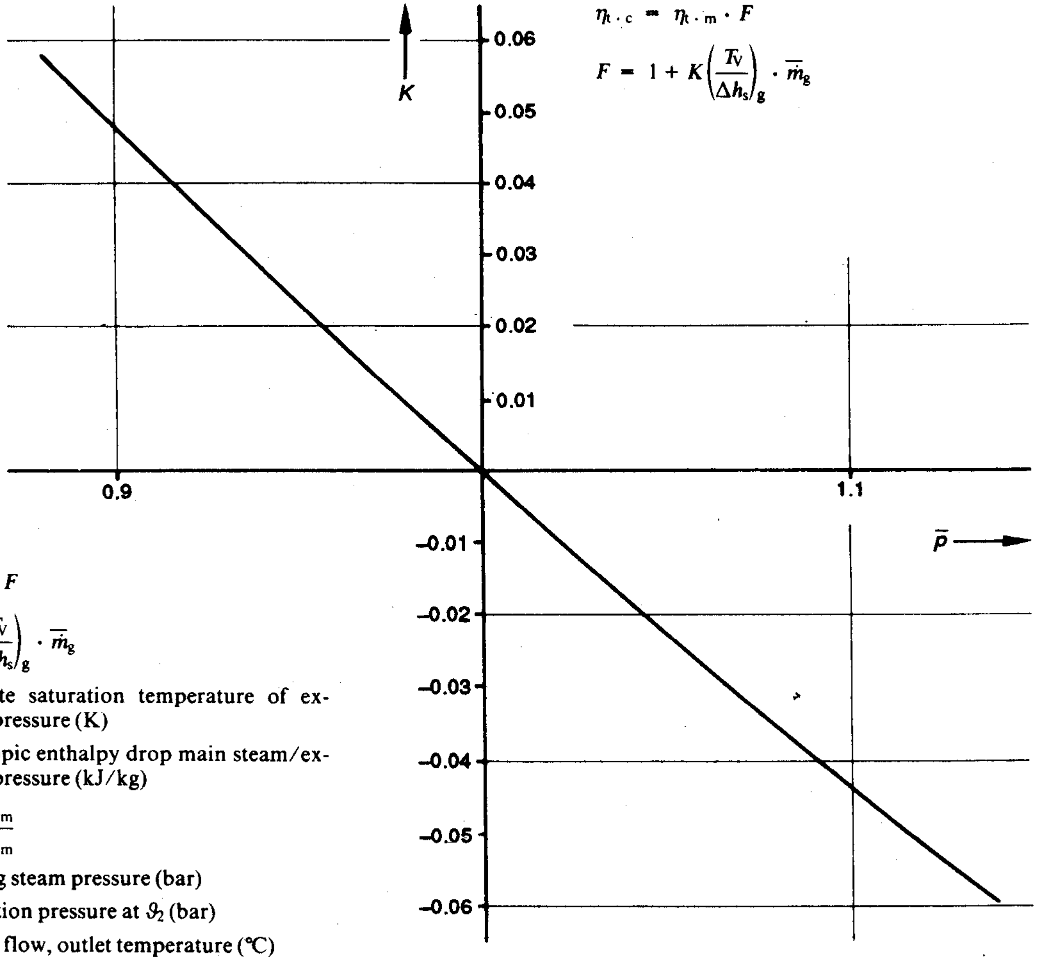
$\bar{m} = \frac{\text{steam flow concerned}}{\text{main steam flow}}$

Valid for: Condensing turbines with regenerative feedwater heating, wet or superheated main steam, with or without reheat, output > 0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.15 \, | F - 1 |$.

FIG. E14. — Correction curve for pressure losses.

Steam in piping, moisture separators, reheaters, bleedings, etc. For main steam side of main steam / steam reheaters, see E8.



$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$

$$F = 1 + K \left(\frac{T_V}{\Delta h_s} \right)_g \cdot \overline{m}_g$$

T_V = absolute saturation temperature of exhaust pressure (K)

Δh_s = isentropic enthalpy drop main steam/exhaust pressure (kJ/kg)

$\overline{p} = \frac{p_{1g} \cdot p_{2m}}{p_{2g} \cdot p_{1m}}$

p_1 = heating steam pressure (bar)

p_2 = saturation pressure at ϑ_2 (bar)

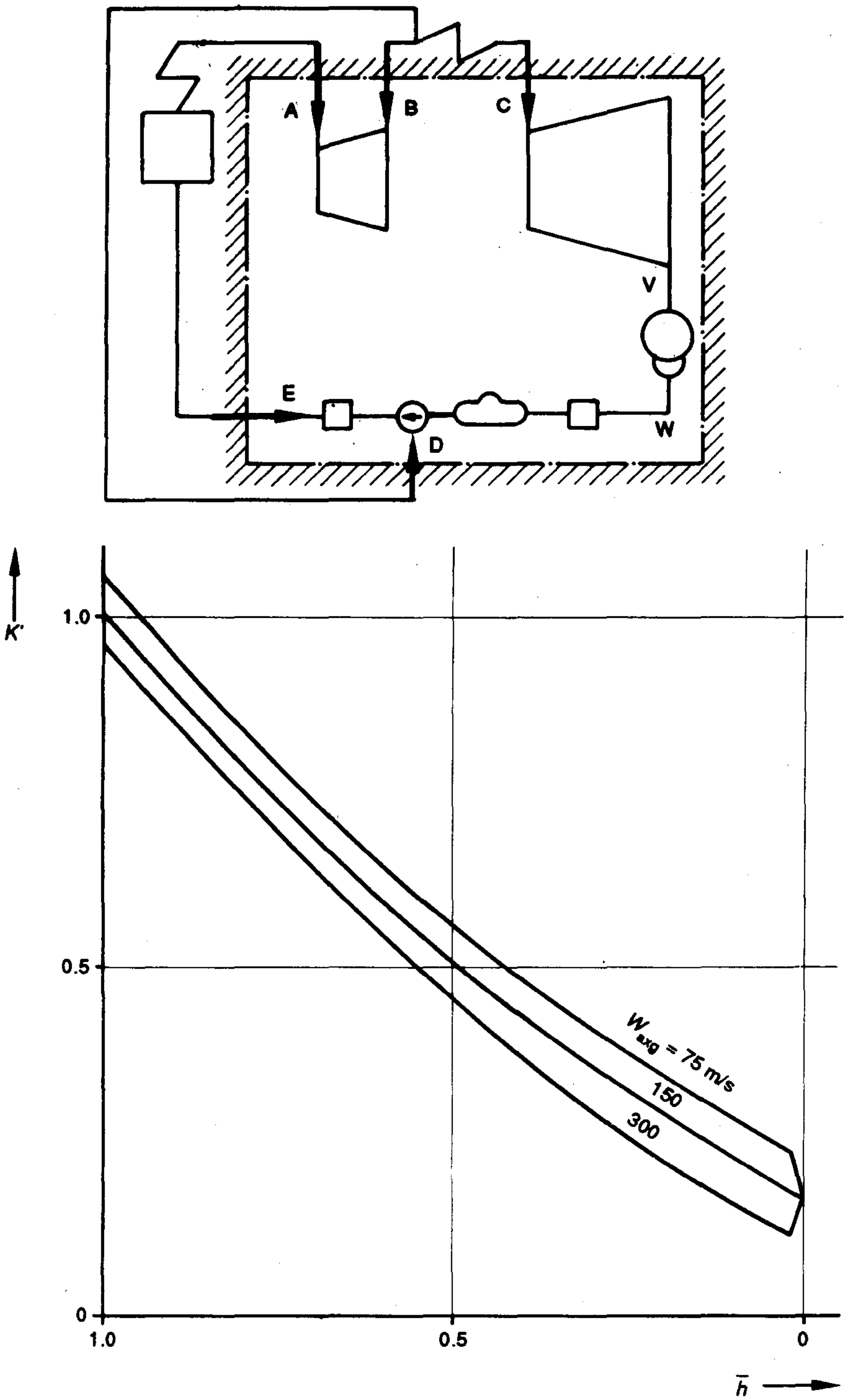
ϑ_2 = heated flow, outlet temperature (°C)

$\overline{m} = \frac{\text{heating steam flow}}{\text{main steam flow}}$

Valid for: Condensing turbines with regenerative feedwater heating, wet or superheated main steam, with or without reheat, output >0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm 0.2 \, | F - 1 |$.

FIG. E15. — Correction curve for terminal differences.
Heaters and bled steam / steam reheater stages.



$\eta_{t\cdot c} = \eta_{t\cdot m} \cdot F$

$F = 1 - \left(Q_A \cdot \frac{\eta_t}{P}\right)_g \cdot \Sigma \left(K \frac{\Delta\bar{\dot{m}}}{1 + \Delta\bar{\dot{m}}}\right)$

Q_A = $\dot{m}_A \cdot h_A$ (kW)

$\dot{m}_A$ = main steam flow excluding main steam flow to steam/steam reheaters if necessary (kg/s)

h_A = main steam enthalpy (kJ/kg)

P = generator output (kW)

$\Delta\bar{\dot{m}} = \left(\frac{\dot{m}}{\dot{m}_A}\right)_m - \left(\frac{\dot{m}}{\dot{m}_A}\right)_g$

$\dot{m}$ = steam subflow (kg/s)

h = $f\,(p,\ s_A)_g$ locus of subflow/main flow linkage, expressed as main flow isentropic enthalpy (kJ/kg)

p = local pressure (bar)

s_A = main steam entropy (kJ/kg K)

$\bar{h} = \left(\frac{h - h_V}{h_A - h_V}\right)_g$

h_V = $f\,(p_V,\ s_A)$ (kJ/kg)

p_V = exhaust pressure (bar)

K = 1.15 K' for reheat turbines in the expansion region behind reheat only

K = K' all others

W_{axg} = last stage axial outlet velocity as guaranteed (m/s)

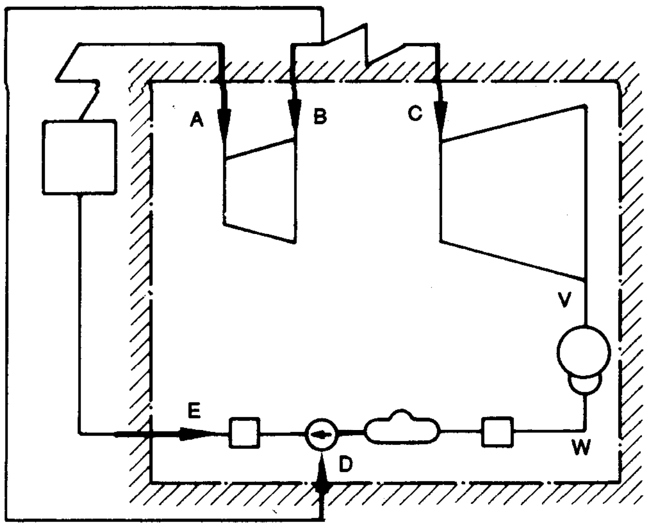
Definition: Subflows are mass flows, which are passing the shaded control area (see figure), but are not directly supplied to or from a primary energy source (boiler, reactor). Therefore $\dot{m}_A$, $\dot{m}_B$, $\dot{m}_C$, $\dot{m}_D$, $\dot{m}_E$ are not subflows. These subflows are particularly leakage flows, extraction flows for non-cycle supply or pump drive, as long as its output is not included in P.

Remarks: Mass flows are inserted in equation (>0) when entering the shaded control area and <0 when leaving it. In case of reheat by main steam, the heat flow to the hp turbine is Q_A, the main steam heat flow to the reheater is ($Q_B + Q_C$).

Valid for: Condensing turbines with regenerative feedwater heating system, main steam wet or superheated, with or without reheat, output >0.2 maximum output, $\dot{m}_B + \dot{m}_C + \dot{m}_D = 0$.

Uncertainty of computation: $\Delta F = \pm\, 0.2\, |F - 1|$.

FIG. E16. — Correction curve for steam subflows.



$\eta_{t \cdot c} = \eta_{t \cdot m} \cdot F$

$F = 1 + \left(Q_E \cdot \frac{\eta_t}{P}\right)_g \cdot \Sigma\,(K \cdot \Delta\bar{\dot{m}})$

$Q_E = \dot{m}_E \cdot h_E$ (kW)

$\dot{m}_E$ = feedwater flow (<0) (kg/s)

h_E = final feedwater enthalpy (kJ/kg)

P = generator output (kW)

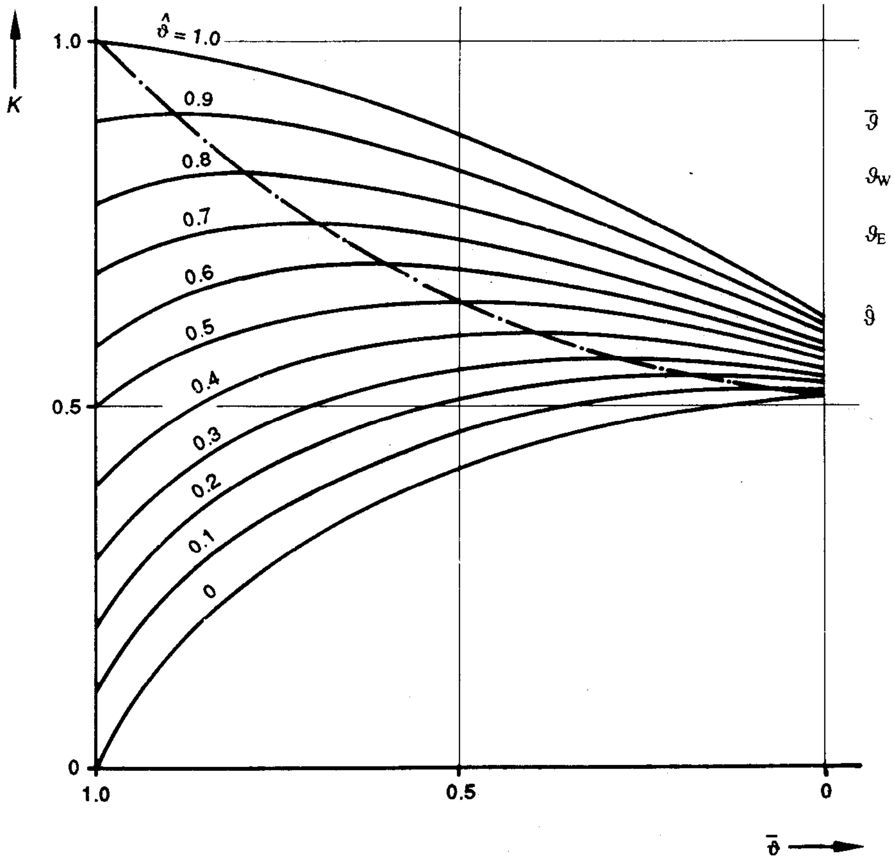
$\Delta\bar{\dot{m}} = \left(\frac{\dot{m}}{\dot{m}_A}\right)_m - \left(\frac{\dot{m}}{\dot{m}_A}\right)_g$

$\dot{m}$ = water subflow (kg/s)

$\dot{m}_A$ = main steam flow, excluding main steam flow to steam/steam reheaters if necessary (kg/s)

ϑ = locus of subflow/main flow linkage, expressed as main flow temperature (°C)

ϑ^* = locus of subflow origin, expressed as local main flow temperature (°C)



$\bar{\vartheta} = \left(\frac{\vartheta - \vartheta_W}{\vartheta_E - \vartheta_W}\right)_g$

ϑ_W = condensate temperature after condenser (°C)

ϑ_E = final feedwater temperature (°C)

$\hat{\vartheta} = \left(\frac{\vartheta^* - \vartheta_W}{\vartheta_E - \vartheta_W}\right)_g$

Definition: Subflows are defined in figure E16. These subflows are particularly leakage flows, flows for non-cycle supply, make-up water and changes in water level.

Remarks: Mass flows are inserted in equations >0 when entering the shaded control area and <0 when leaving it. Complete or partial bypassing of heaters will be simulated by two equal subflows of same subflow temperature, but different direction and different mainflow temperature. Correction for enthalpy rise in feedpump will be simulated by feedflow as subflows leaving the system and entering again with different $\hat{\vartheta}$ but same $\bar{\vartheta}$.

Valid for: Condensing turbines with regenerative feedwater heating, wet or superheated main steam, with or without reheat, output >0.2 maximum output.

Uncertainty of computation: $\Delta F = \pm\, 0.2\,|F - 1|$.

FIG. E17. — Correction curve for water subflows and enthalpy rise in feedpump.

TABLE EI

Mathematical formulations representing curves E1 to E17

E1: $K' = -0.455 \ln \overline{\dot{m}}_A$

E2: $$K = 0.0235 \left[\left(\ln \left(\overline{\dot{m}}_A \frac{W_{axg}}{150} \right) \right)^2 - \left(\ln \frac{W_{axg}}{150} \right)^2 \right]$$

E3: $$K = -\sqrt{\frac{S_{max}}{2\,000\,000}} \cdot \left(\frac{0.005}{\bar{p} \cdot \sqrt{\dot{m}_A}} + 0.0075\, \bar{p} \cdot \sqrt{\dot{m}_A} \right) + \frac{0.012}{\bar{p} \cdot \sqrt{\dot{m}_A}} - 0.003\, p \cdot \sqrt{\dot{m}_A}$$

E4: $$K = 0.0186 \cdot \sqrt{\bar{S}} \ln \frac{\cos \varphi_g}{\cos \varphi_m}$$

E5: $$K = 0.3888 \cdot v \left(\frac{\overline{\dot{m}}_A}{\bar{p}_A} - 1 \right) + \frac{1}{426 - 444 \dfrac{\overline{\dot{m}}_A\, v}{\bar{p}_A}} - \frac{1}{426 - 444\, v}$$

$$K' = 0.455 \cdot \ln \frac{\bar{p}_A}{\overline{\dot{m}}_A}$$

E6: *A:* $K = -0.348 \ln \bar{j}$
B: $K = -0.178 \ln \bar{j}$
C: $K = -0.144 \ln \bar{j}$

E7: See Figure E7

E8: $$K = 0.016 \left(\frac{p_2}{p_A} \right)_g (\bar{p} - 1); \qquad K' = 0.0035 \ln \bar{p} \cdot \ln \frac{W_{axg}}{150}; \qquad K'' = 0.006 \ln \bar{p} \ln \bar{P}$$

E9: $$K = [y_1]^2 - [y_2]^2 - 0.03 \ln \bar{p}$$

$$y_1' = \ln \left[\left(\frac{380}{\overline{\dot{m}}_A\, W_{axg}} \right)^{1.1} \bar{p} \right] \qquad y_1 = y_1' \text{ for } y_1' > 0;\ y_1 = 0 \text{ for } y_1' \leqslant 0$$

$$y_2' = \ln \left[\left(\frac{380}{\overline{\dot{m}}_A\, W_{axg}} \right)^{1.1} \right] \qquad y_2 = y_2' \text{ for } y_2' > 0;\ y_2 = 0 \text{ for } y_2' \leqslant 0$$

E10: See figure E10

E11: $$K = 0.0001 \ln \left(\frac{p_1}{p_v} \cdot \frac{150}{W_{ax}} \right)_g$$

E12: $$K = 0.30 \left[1 - \frac{z+1}{z} (\bar{\vartheta} + 0.5\, \Delta\bar{\vartheta} + 0.03) \right]$$

$$E13:\ K = 0.02 + 0.1 \ln \bar{p} + 0.067 \ln \frac{W_{\mathrm{axg}}}{150}$$

$$E14:\ K = -0.455 \ln \bar{p}$$

$$E15:\ K = -0.455 \ln \bar{p}$$

$$E16:\ K' = -0.16 + 1.16\,\bar{h} + 0.32\,(\bar{h} - 1)^2 - \left(0.058 \ln \frac{W_{\mathrm{axg}}}{150}\right)(1.5 - 0.5\,\bar{h})$$

$$E17:\ K = (1 - (1 - \hat{\vartheta})^{0.95})\,(1 - 0.38\,(1 - \bar{\vartheta})^{1.8}) + (1 - \hat{\vartheta})^{0.95}\,[(1 - \bar{\vartheta})^{0.75} - 0.5\,(1 - \bar{\vartheta})^{1.5}]$$
$$+ \left(\frac{0.0001}{(\bar{\vartheta} - \hat{\vartheta})^2 + 0.01} - \frac{0.0001}{1.01}\right)(1 - \hat{\vartheta})^{0.3}$$

APPENDIX F

SHORT-STATISTICAL DEFINITION OF MEASURING UNCERTAINTY AND ERROR PROPAGATION IN ACCEPTANCE TESTS

(see 7.1 and 7.5.1)

The measurement of a physical quantity can be carried out with limited accuracy only, because of the inevitable measuring error

$$\text{error} = \text{measured value} - \text{true value}$$

In practice, the relative error is used very often, where

$$\text{relative error} = \frac{\text{error}}{\text{true value}} \approx \frac{\text{error}}{\text{measured value}}$$

If the errors are randomly distributed, the accuracy of the determination of a constant quantity (length, weight) can be increased by repeating the measurement and calculating the arithmetic mean value. The deviations of the individual measured values from the mean will follow the Gaussian Law (no systematic error present). From the standard deviation of the resulting Gauss curve, the confidence limits V_{x_i} of the error of an individual measurement x_i can be calculated.

The statement

$$V_{x_i} = \pm 2\delta \text{ for } P \approx 95\%$$

means that the measured value x_i is situated with a statistical probability of 95% in the range

$$x \pm 2\delta$$

where x is the true value.

V_{x_i} are the *confidence limits of the error* for the value x_i.

The error of the measured value x_i with respect to the unknown true value x is not greater than $\pm V_{x_i}$ with a probability of 95%.

The confidence limits V_x for a statistical probability of 95% are called *measuring uncertainty* in the following. The *relative measuring uncertainty* is defined by

$$\tau_x = \frac{V_x}{x} \sim \frac{V_x}{x_i}$$

For a direct application of the statistical theory of uncertainty of measurements, a great number of measurements of a constant physical quantity with randomly distributed statistically independent errors is necessary, as indicated above. This condition is not fulfilled for a thermal acceptance test.

In order to allow statements of the probable uncertainty of values measured during thermal acceptance tests and to permit the application of the laws of error propagation for the determination of the uncertainty of the results calculated from these values, the following generalizations of the definitions are introduced.

In thermal acceptance tests the quantities to be measured are not constant in time. A series of readings is taken at regular intervals during the test period and the arithmetic mean value is formed. This mean value contains three different types of independent errors:

a) Random reading error with confidence limit V_A[1].

b) Integration error with confidence limit V_I.

c) Non-detectable systematic error[2] of the measuring equipment with confidence limit V_S or accuracy class G[3].

The confidence limit (measuring uncertainty) of the averaged reading value $\bar{x}$ is:

$$V_{\bar{x}} = \pm\sqrt{V_A^2 + V_I^2 + V_S^2}\,^{4)}$$

In normal acceptance test practice, the number of readings is sufficiently large to render V_A and V_I negligible, so that V_S or G is significant for the measuring uncertainty.

The average value $\bar{x}$ with its confidence limits of the error $V_{\bar{x}}$ can be interpreted as an individual measurement according to statistical theory.

The confidence limits V_S cannot be established from a series of measurements of a time-fluctuating variable with one and the same instrumentation by statistical methods and have to be determined by different means for the different kinds of measured values. They shall always be referred to a statistical probability $P \approx 95\%$.

The results (e.g. mass flow, thermal efficiency, etc.) are calculated from measured variables as well as from physical properties and coefficients. Generally the result y is derived from:

n_x measured variables x_j and from
n_b physical properties b_l and
n_c coefficients c_m:

$$y = F(x_j, b_l, c_m)$$

V_b and V_c are the confidence limits of the errors of the physical properties and coefficients due to non-detectable systematic errors for $P \approx 95\%$.

If all measured values, physical property values and coefficients and their confidence limits of the error (measuring uncertainty) are independent of each other and determined for the same probability P (e.g., 95%), they can be added according to the law of error propagation (square root rule) and the measuring uncertainty V_y of the result determined in this manner will have the same probability. It is:

$$V_y = \pm\sqrt{\sum_{j=1}^{n_x}\left(\frac{\partial y}{\partial x_j} V_{xj}\right)^2 + \sum_{l=1}^{n_b}\left(\frac{\partial y}{\partial b_l} V_{bl}\right)^2 + \sum_{m=1}^{n_c}\left(\frac{\partial y}{\partial c_m} V_{c_m}\right)^2}$$

where V_{x_j} is the measuring uncertainty of an individual variable x_j.

The calculation methods applied in this appendix, derived from the probability theory, are based on the assumption that all the values used in the calculation are statistically absolutely independent of each other.

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[1] A systematic reading error is equivalent to a non-detectable or non-detected systematic error of the measuring equipment and is to be treated accordingly under[3].

[2] The non-detectable systematic error is largely constant in one and the same acceptance test measurement, but are assumed to be randomly distributed in several tests using the same measuring equipment. As a result, the equations of probability theory may be used.

[3] For several types of instruments, the manufacturer indicates the accuracy class G, which is a confidence limit of the error with a probability P = 100%. For application, the reference value of G, mostly the scale end value, and limitations of operating conditions have to be closely observed.

[4] Confidence limits (measuring uncertainty) which are independent of each other and are defined for the same probability P can be added according to the law of error propagation (square root rule).

In most cases, however, physical properties and coefficients depend on the measured variables. In such cases the natural relationship between measured variables and physical properties and coefficients:

$$b_l = F_b(x_j)$$
$$c_m = F_c(x_j)$$

shall be introduced into the equation for y prior to application of the error propagation law.

In many cases, the measuring uncertainties of measured values are not independent of these values themselves, or the uncertainties of values measured in parallel or with the same instrumentation are mutually interdependent. The error propagation law as shown above is then not applicable or has to be applied differently.

Indications for the treatment of the more common cases of interdependence are given in the following.

In view of the many influencing factors involved, the measuring uncertainty of all pressure difference devices for mass flows, as well as all thermometers or thermocouples, shall be considered to be independent. Measurements with instruments which have been calibrated on the same test bed (e.g., orifices and nozzles calibrated together with the test section as well as thermometers and thermocouples) shall be considered to be dependent on each other in view of the calibration error.

Pressure difference devices of the same dimensions working under practically the same conditions (e.g. measurement of initial steam flow in several parallel flows) shall all be considered to be entirely dependent on each other.

For the dependence of the measuring uncertainty of instrument transformers, see 7.3.1. For the evaluation of the test, it is recommended that deviations from the assumption of independence of the variables should not be considered if a considerable simplification of the evaluation can be achieved without any appreciable effect on the results, particularly on the measuring uncertainty.

The interdependence due to instruments being calibrated on common test beds is generally negligible if the error limits of calibration are small compared with the other measuring uncertainty of the measurement. This also applies to the use of one precision potentiometer for all temperature measurements.

To allow correctly for all the interdependences in calculating the measuring uncertainty of the results generally involves a highly sophisticated calculation process and is impracticable in most cases. It is recommended to accept deviations from the principle of independence of measured values and their uncertainties, if a considerable simplification of the computation can thus be achieved without appreciable modification of the results and their uncertainties.

APPENDIX G

CALCULATION OF MEASURING UNCERTAINTY OF OUTPUT – ELECTRICAL MEASUREMENT (see 7.3.1)

G1. Introduction

In general the law of error propagation with geometrical summation of the individual measurement uncertainties is also applied for polyphase measurements (two-wattmeter or three-wattmeter method). Practical experiments have shown that the transforming ratio errors, as well as the phase-angle errors of the transformers in the various phases, are not fully independent. This also applies to the calibration of transformers.

Therefore, the summation of errors has to be separated into a geometrical part and an arithmetical part.

In the calculation of the measurement uncertainty of the total power as the sum of the power in the individual phases, the following measurement uncertainties have to be taken into account.

a) Wattmeter and watthour meter

The summation of the measurement uncertainty due to the wattmeters or watthour meters in the various phases is geometrical.

b) Transforming errors of voltage transformers

The summation of the measurement uncertainties due to the transforming errors of voltage transformers between the various phases is arithmetical.

c) Phase-angle errors of voltage transformers

The summation of the measurement uncertainties due to the phase-angle errors of voltage transformers is arithmetical.

d) Transforming errors of current transformers

The summation of the measurement uncertainty due to the transforming errors of current transformers in the various phases is arithmetical.

e) Phase-angle errors of current transformers

The summation of the measurement uncertainty due to the phase-angle errors of current transformers is arithmetical.

f) Total error

The uncertainty in total power is calculated by geometrical summation of the uncertainties mentioned under *a), b), c), d)* and *e).*

Notes 1. — The summation mentioned under *c)* or *e)* is sometimes made geometrically. This is not correct in the philosophy of this appendix, as the statement in the first paragraph applies to the phase angle errors as well.

2. — In the formulae an uncertainty for the temperature influence on wattmeters or watthour meters is introduced. The temperature factor should be indicated by the manufacturer of the wattmeters/watthour meters.

If the influence of the temperature on the wattmeter reading is exactly known as a function of the temperature and this has been applied as a correction in the evaluation of the measured power, this influence on the measurement uncertainty can be eliminated in the formula for measurement uncertainty.

G2. Formulae

Formulae are presented for the application of wattmeters, single-phase watthour meters and three-phase watthour meters for the following cases, both for three-phase four wire circuits (three-wattmeter method) and for three-phase three-wire circuits (two-wattmeter method or Aron arrangement):

a) calibrated transformers and wattmeters/watthour meters;

b) calibrated transformers and non-calibrated wattmeter/watthour meter(s) with specified accuracy class;

c) non-calibrated transformers with specified accuracy class and calibrated wattmeters/watthour meter(s);

d) non-calibrated transformers and wattmeters/watthour meter(s) each with specified accuracy class.

In the formulae below the terms with r_w and r_{wh} can be ignored, and the following symbols are used:

c_u = calibration inaccuracy in the ratio error of the voltage transformers (%)
c_i = calibration inaccuracy in the ratio error of the current transformers (%)
$\delta_{\varphi u}$ = calibration inaccuracy in the phase angle of the voltage transformers in mrad
$\delta_{\varphi i}$ = calibration inaccuracy in the phase angle of the current transformers in mrad
c_w = calibration inaccuracy of the wattmeters in scale divisions
c_{wh} = calibration inaccuracy of the watthour meters (%)
r_w = reading inaccuracy of the wattmeters in scale divisions
r_{wh} = reading inaccuracy of the watthour meters (%)
G_u = accuracy class of the voltage transformers (%)
G_i = accuracy class of the current transformers (%)
G_w = accuracy class of the wattmeters in % of full scale
G_{wh} = accuracy class of the watthour meters (%)
$D_{\varphi u}$ = class accuracy for phase angle of the voltage transformers in mrad
$D_{\varphi i}$ = class accuracy for phase angle of the current transformers in mrad
α_E = full scale of the wattmeters
α_1, α_2 = actual wattmeter/watthour meter reading (two-wattmeter/watthourmeter method)
$\bar{\alpha}$ = mean value of the wattmeter readings (three-wattmeter method)
t = temperature in degrees Celsius
λ = temperature coefficient of wattmeter/watthourmeter in % accuracy/K
V_p = confidence limit of the measuring uncertainty

$$K = \frac{\alpha_2}{\alpha_1 + \alpha_2}$$

I — *Three-phase four-wire circuit*

Case a:

*a.*1 *Single-phase wattmeters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \frac{\left(\frac{c_w}{\bar{\alpha}} \cdot 100\right)^2 + \left(\frac{r_w}{\bar{\alpha}} \cdot 100\right)^2}{3} + 1/3 \cdot \{\lambda (t - 23)\}^2$$
$$+ \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*a.*2 *Single-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \frac{(c_{wh})^2 + (r_{wh})^2}{3} + 1/3\, \lambda^2 (t - 23)^2 + \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*a.*3 *Three-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + (c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t - 23)^2 + \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

Case b:

*b.*1 *Single-phase wattmeters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \frac{G_w^2 \cdot \left(\frac{\alpha_E}{\alpha}\right)^2 + \{\lambda(t-23)\}^2 + \left(\frac{r_w}{\alpha} \cdot 100\right)^2}{3}$$

$$+ \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*b.*2 *Single-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \frac{(G_{wh})^2 + \lambda^2 (t-23)^2 + (r_{wh})^2}{3} + \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*b.*3 *Three-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + (G_{wh})^2 + \lambda^2 (t-23)^2 + (r_{wh})^2 + \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

In general, the calibration uncertainty figures c and δ* of the transformers and the uncertainties in reading r_w and r_{wh} of the wattmeters/watthour meters can be ignored as compared with the accuracy class of the wattmeters/watthour meters. The formulae may then be simplified to:

— *for single-phase wattmeters:* $V_p^2 = \frac{1}{3}\left\{G_w^2 \left(\frac{\alpha_E}{\alpha}\right)^2 + \lambda^2 (t-23)^2\right\}$

— *for single-phase watthour meters:* $V_p^2 = \frac{1}{3}\left\{G_{wh}^2 + \lambda^2 (t-23)^2\right\}$

— *for three-phase watthour meters:* $V_p^2 = G_{wh}^2 + \lambda^2 (t-23)^2$

Case c:

*c.*1 *Single-phase wattmeters*

$$V_p^2 = (G_i)^2 + (G_u)^2 + \frac{\left(\frac{c_w}{\alpha} \cdot 100\right)^2 + \left(\frac{r_w}{\alpha} \cdot 100\right)^2}{3}$$

$$+ 1/3\, \lambda^2 (t-23) + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*c.*2 *Single-phase watthour metres*

$$V_p^2 = (G_i)^2 + (G_u)^2 + \frac{(c_{wh})^2 + (r_{wh})^2}{3}$$

$$+ 1/3\, \lambda^2 (t-23)^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

*c.*3 *Three-phase watthour metres*

$$V_p^2 = (G_i)^2 + (G_u)^2 + (c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t-23)^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi$$

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* For usual power factors.

The use of the value of the class accuracy of the transformers per phase in the accuracy formulas for total power is in fact based upon the arithmetical addition of absolute uncertainties.

$$(G_i = G_{i1} = G_{i2} = G_{i3}; \text{the same applies to } G_u, D_i \text{ and } D_u)$$

Ignoring the calibration and reading inaccuracies *c* and *r* of the wattmeter leads to the simplified formulae valid for cases *c*.1, *c*.2 and *c*.3, respectively:

— *for single-phase wattmeters:*

$$V_p^2 = G_i^2 + G_u^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi + \frac{1}{3} \cdot \lambda^2 (t - 23)^2$$

— *for single-phase watthour wattmeters:*

$$V_p^2 = G_i^2 + G_u^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi + \frac{1}{3} \cdot \lambda^2 (t - 23)^2$$

— *for three-phase watthour wattmeters:*

$$V_p^2 = G_i^2 + G_u^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi + \lambda^2 (t - 23)^2$$

Case d:

d.1 *Single-phase wattmeters*

$$V_p^2 = (G_i)^2 + (G_u)^2 + \frac{G_w^2 \cdot \left(\frac{\alpha_E}{\alpha}\right)^2 + \lambda^2 (t - 23)^2 + \left(\frac{r_w}{\alpha} \cdot 100\right)^2}{3}$$

$$+ \{(D_i)^2 + (D_u)^2\} \cdot 10^{-2} \tan^2 \varphi$$

d.2 *Single-phase watthour meters* (one phase)

$$V_p^2 = (G_i)^2 + (G_u)^2 + \frac{(G_{wh})^2 + \lambda^2 (t - 23)^2 + (r_{wh})^2}{3} + \{(D_{\varphi i})^2 + (D_{\varphi u})^2 \cdot 10^{-2} \tan^2 \varphi\}$$

d.3 *Three-phase watthour meters* (three phase)

$$V_p^2 = (G_i)^2 + (G_u)^2 + (G_{wh})^2 + \lambda^2 (t - 23)^2 + (r_{wh})^2 + \{(D_{\varphi i})^2 + (D_{\varphi u})^2 \cdot 10^{-2} \tan^2 \varphi\}$$

In the formulae, the reading uncertainties r_w and r_{wh} may be ignored as compared with the class accuracies.

II — *Three-phase three-wire circuit*

It is assumed that the voltage transformers are connected in open-V.

The formulae valid for the cases *d*.1, *d*.2 and *d*.3, respectively, are also available in the case of a Y connection, a DZo connection or a Dyll connection of the voltage transformers if they have the same accuracy class, when they are not calibrated or when the voltage transformers are calibrated by means of the same standard voltage transformer and calibration bridge; in the opposite case the errors to be applied in the formulae for a three-phase three-wire circuit should be calculated by means of the following formulae in the case of a Y connection.

1) In the case of calibrated voltage transformers

$$(c_u)_{RS} \approx \frac{1}{2}\{(c_u)_R + (c_u)_S\} + 2.9 \cdot 10^{-2}\{(\delta_{\varphi u})_R - (\delta_{\varphi u})_S\}$$

$$(\delta_{\varphi u})_{RS} \approx \frac{1}{2}\{(\delta_{\varphi u})_R + (\delta_{\varphi u})_S\} - 2.9\{(c_u)_R - (c_u)_S\}$$

and

$$(c_u)_{TS} \approx \frac{1}{2}\{(c_u)_T + (c_u)_S\} - 2.9 \cdot 10^{-2}\{(\delta_{\varphi u})_T - (\delta_{\varphi u})_S\}$$

$$(\delta_{\varphi u})_{TS} \approx \frac{1}{2}\{(\delta_{\varphi u})_T + (\delta_{\varphi u})_S\} + 2.9\{(c_u)_T - (c_u)_S\}$$

2) In the case of non-calibrated voltage transformers

$$(G_u)_{RS} \approx \frac{1}{2}\{(G_u)_R + (G_u)_S\} + 2.9 \cdot 10^{-2}\{(D_{\varphi u})_R - (D_{\varphi u})_S\}$$

$$(D_{\varphi u})_{RS} \approx \frac{1}{2}\{(D_{\varphi u})_R + (D_{\varphi u})_S\} - 2.9\{(G_u)_R - (G_u)_S\}$$

and

$$(G_u)_{TS} \approx \frac{1}{2}\{(G_u)_T + (G_u)_S\} - 2.9 \cdot 10^{-2}\{(D_{\varphi u})_T - (D_{\varphi u})_S\}$$

$$(D_{\varphi u})_{TS} \approx \frac{1}{2}\{(D_{\varphi u})_T + (D_{\varphi u})_S\} + 2.9\{(G_u)_T - (G_u)_S\}$$

Attention has to be paid to the symmetry of the voltages over the transformers

Case a:

*a.*1 *Single-phase wattmeters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \left\{\left(\frac{c_w}{\alpha_i} \cdot 100\right)^2 + \left(\frac{r_w}{\alpha_i} \cdot 100\right)^2\right\}(1 - K)^2$$

$$+ \left\{\left(\frac{c_w}{\alpha_2} \cdot 100\right)^2 + \left(\frac{r_w}{\alpha_2} \cdot 100\right)^2\right\} K^2 + \lambda^2 (t - 23)^2 \{(1 - K)^2 + K^2\}$$

$$+ 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\} \text{ with } K = \frac{\alpha_2}{\alpha_1 + \alpha_2}$$

*a.*2 *Single-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \{(c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t - 23)^2\}$$
$$x\{(1 - K)^2 + K^2\} + 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\}$$

*a.*3 *Three-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + (c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t - 23)^2 + 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\}$$

Case b:

b.1 *Single-phase wattmeters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + (1 - K)^2 \left\{ G_w^2 \left(\frac{\alpha_E}{\alpha_2}\right)^2 + \lambda^2 (t - 23))\}^2 \right\}$$

$$+ K^2 \left\{ (G_w)^2 + \left(\frac{\alpha_E}{\alpha_2}\right)^2 + \lambda^2 (t - 23)^2 \right\} + \left(\frac{r_w}{\alpha_1} \cdot 100\right)^2 (1 - K)^2$$

$$+ \left(\frac{r_w}{\alpha_2} \cdot 100\right) K^2 + 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\}$$

b.2 *Single-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + \{(G_{wh})^2 + \lambda^2 (t - 23)^2\} \cdot \{(1 - K)^2 + K^2\}$$

$$+ (r_{wh})^2 \{(1 - K)^2 + K^2\} + 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\}$$

b.3 *Three-phase watthour meters*

$$V_p^2 = (c_i)^2 + (c_u)^2 + (G_{wh})^2 + \lambda^2 (t - 23)^2 + (r_{wh})^2$$

$$+ 3 \cdot 10^{-2} (1 - 2K)^2 \{(\delta_{\varphi i})^2 + (\delta_{\varphi u})^2\}$$

In general, the calibration uncertainty figures c and δ* of the transformers and the uncertainties in reading r_w and r_{wh} of the wattmeters/watthour meters can be ignored as compared with the accuracy class of the wattmeters/watthour meters.

The formulae valid for cases *b*.1, *b*.2 and *b*.3, respectively, may then be simplified to:

— *for single-phase wattmeters:*

$$V_p^2 = G_w^2 \left\{ (1 - K)^2 \left(\frac{\alpha_E}{\alpha_1}\right)^2 + K^2 \left(\frac{\alpha_E}{\alpha_2}\right)^2 \right\} + \{(1 - K)^2 + K^2\} \lambda^2 (t - 23)^2$$

— *for single-phase watthour meters:*

$$V_p^2 = \left\{ (G_{wh})^2 + \lambda^2 (t - 23)^2 \right\} \cdot \{(1 - K)^2 + K^2\}$$

— *for three-phase watthour meters:*

$$V_p^2 = (G_{wh})^2 + \lambda^2 (t - 23)^2$$

Case c:

c.1 *Single-phase wattmeters*

$$V_p^2 = G_i^2 + G_u^2 + \left\{ \left(\frac{c_w}{\alpha_1} \cdot 100\right)^2 + \left(\frac{r_w}{\alpha_1} \cdot 100\right)^2 \right\} \cdot (1 - K)^2$$

$$+ \left\{ \left(\frac{c_w}{\alpha_2} \cdot 100\right)^2 \right\} + \left(\frac{r_w}{\alpha_2} \cdot 100\right)^2 K^2 + \lambda^2 (t - 23)^2 \cdot \{(1 - K)^2 + K^2\}$$

$$+ 3 \cdot 10^{-2} (1 - 2\,K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\}$$

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* For usual power factors.

*c.*2 *Single-phase watthour meters*

$$\begin{aligned} V_p^2 = {} & G_i^2 + G_u^2 + (c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t-23)^2 \cdot \{(1-K)^2 + K^2\} \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \cdot 10^{-2} \tan^2 \varphi \end{aligned}$$

*c.*3 *Three-phase watthour meters*

$$\begin{aligned} V_p^2 = {} & (G_i)^2 + (G_u)^2 + (c_{wh})^2 + (r_{wh})^2 + \lambda^2 (t-23)^2 \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \end{aligned}$$

Ignoring the calibration and reading inaccuracies of the wattmeters/watthour meters, the apropriate formulae for cases *c.*1, *c.*2 and *c.*3 can simplified to:

- *for wattmeters:*

$$\begin{aligned} V_p^2 = {} & G_i^2 + G_u^2 + \lambda^2 (t-23)^2 \cdot \{(1-K)^2 + K^2\} \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \end{aligned}$$

- *for single-phase watthour meters:*

$$\begin{aligned} V_p^2 = {} & G_i^2 + G_u^2 + \{\lambda^2 (t-23)\}^2 \{(1-K)^2 + K^2\} \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \end{aligned}$$

- *for three-phase watthour meters:*

$$V_p^2 = G_i^2 + G_u^2 + \lambda^2 (t-23)^2 + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\}$$

Case d:

*d.*1 *Single-phase wattmeters*

$$\begin{aligned} V_p^2 = {} & G_i^2 + G_u^2 + (1-K)^2 \left\{ (G_w)^2 \left(\frac{\alpha_E}{\alpha_1} \right)^2 + \lambda^2 (t-23)^2 \right\} + K^2 \left\{ (G_w)^2 \left(\frac{\alpha_E}{\alpha_2} \right)^2 \right. \\ & \left. + \lambda^2 (t-23)^2 \right\} + \left(\frac{r_w}{\alpha_1} \cdot 100 \right)^2 (1-K)^2 + \left(\frac{r_w}{\alpha_2} \cdot 100 \right)^2 K^2 \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \end{aligned}$$

*d.*2 *Single-phase watthour meters*

$$\begin{aligned} V_p^2 = {} & G_i^2 + G_u^2 + \{(G_{wh})^2 + \lambda^2 (t-23)^2\} \cdot \{(1-K)^2 + K^2\} + (r_{wh})^2 \{(1-K)^2 + K^2\} \\ & + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\} \end{aligned}$$

*d.*3 *Three-phase watthour meters*

$$V_p^2 = G_i^2 + G_u^2 + (G_{wh})^2 + \lambda^2 (t-23)^2 + (r_{wh})^2 + 3 \cdot 10^{-2} (1-2K)^2 \{(D_{\varphi i})^2 + (D_{\varphi u})^2\}$$

In these formulae the terms with r_w and r_{wh} can be ignored.